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Trade Associations

PEOPLE are naturally suspicious of trade associations. They see in them combinations in restraint of trade, convoked for the purpose of hiking up prices and mutual support in the making of "unconscionable profits."

But as industry is conducted today, trade organization is inevitable and is becoming more and more desirable.

There are many things that manufacturers of the same class can legitimately discuss to their mutual advantage and to that of their customers.

It is a great deal better for the customer that these men should be conspiring as to how they can improve their output, make their apparatus more efficient and more nicely adapted to the customer's purposes and combining their forces in friendly effort for the improvement of their collective status through greater service, than that they should be warring upon one another.

We have no toleration for the various forms of gentlemen's agreement by which more than a fair profit is extorted from the customer, and it must not be forgotten that it is the ultimate consumer that pays that profit in the end.

But the kind of competition that leads to inferior product and unsatisfactory service is likely to prove costly to the purchaser even at a reduced price.

There are trade associations composed of high-minded public-spirited men who can see the import of their business as a factor in the nation's industry and the people's welfare, who realize that it will find its greatest growth and firmest establishment in integrity of product and efficiency of service, and who are big enough to sink competitive animosities in a united effort to realize its greatest collective possibilities.

And such associations, kept free from perversion to selfish and unwarrantable ends, are useful factors in the organization and systemization of industry for its legitimate object, the efficient adaptation of the materials and products of nature to the purposes of man.

F. R. Low

Suggestions for Saving Heat in Manufacturing Plants

By CHARLES L. HUBBARD*

WHILE the scarcity and high price of fuel have brought out many suggestions for the elimination or reduction of heat losses in party and industrial plants, a glance over the manufacturing section of any large city on a cold morning shows that the subject of exhaust utilization has not yet reached a stage where it can be profitably dropped even if there is practically nothing new to be offered. This article outlines several methods for reclaiming much of the heat that is now ordinarily wasted in manufacturing plants, and the suggestions here presented should furnish much food for thought to the engineer operating plants of this character.

THE points to be considered in the present article are not original with the writer and, for the most part, have already been mentioned in connection with isolated cases. In view of the wider application possible, both in new plants and those already installed, and the saving in fuel that may be made under favorable conditions, a brief discussion of of these matters may be well worth while at the present time.

The item to be considered is the loss of the heat that passes away in the cooling water from a condensing engine or turbine. While this loss is not so spectacular as that from an atmospheric exhaust pipe, it is from 75 to 80 per cent as much, under average conditions, as that from the same engine operating non-condensing and exhausting to the atmosphere. Where a considerable part of the exhaust steam can be utilized for heating purposes of various kinds, the usual arrangement is to employ a non-condensing engine or turbine, supplemented, perhaps, by a condensing outfit for use during the non-heating season. If, however, the radiation in the various buildings may be made to serve as an air-cooled condenser, we not only save the heat in the exhaust stream, but also obtain the higher engine economy possible by running condensing. This arrangement is shown in diagram in Fig. 1 and is easily made in case of any two-pipe system of steam heating. If the return-line vacuum system is already installed, it simply means a cross-connection between the main return line and the condenser.

With a gravity system automatic return valves must be placed on all radiators and coils and a vacuum pump installed for use when the condenser is not running. But if the present system is operating on a gravity return of condensation to a receiving tank, no vacuum pump will be required, as the old arrangement may be made use of when the condenser is out of service. In general, it is better in new work to furnish a vacuum pump for relay work on account of the long runs of return piping usually found in manufacturing plants.

Referring to Fig. 1, it will be seen that the combination system simply consists in the exhaust steam passing to the radiation, which serves as an air-cooled surface condenser, instead of to the condenser proper. Suction on the return, for removing the condensation, is produced by connecting the return main with the engine exhaust pipe close to the condensing chamber, as indicated in the drawing. If the condensing capacity of the radiating system was always sufficient to care for all of the exhaust steam, an ordinary vacuum pump for removing the air and condensation from the return system would answer all purposes, and the condenser might be dispensed with. Under practical working conditions the quantity of steam required by the heating system varies widely, so that the condenser must be at hand to care automatically for any surplus and maintain a proper vacuum on the engine or turbine. In operation the back pressure valve *A* is set to open at ½ to 1 lb. gage and the relief valve *B* weighted to hold about 20 in. of vacuum on the engine. The full vacuum of 26 to 28 in. is put on the condenser. In case all the exhaust steam cannot be condensed in the radiating system and the back pressure on the engine rises slightly, valve *B* will open and the surplus steam pass directly into the condenser, upon which the full vacuum is maintained at all times. As the season advances and less steam is required, more and more of the exhaust will pass through valve *B* until heat is no longer required on the buildings and gate valve *C* is closed. In case there is not enough exhaust steam to maintain the normal working pressure in the heating system (20 in. vacuum, or 5 lb. absolute pressure), the pressure reducing valve *D* automatically admits the required amount of live steam to make up the deficiency.

Of course, if all of the exhaust steam from a non-condensing engine could be utilized throughout the year, a condensing plant would not be necessary for reducing the steam consumption of the engine, and then making up the deficiency thus caused by drawing live steam from the boilers. As a matter of fact this condition practically never occurs, and the varying proportions of steam available and steam required for each day of the entire year must be taken into account when deciding upon the best arrangement to be used.

Another scheme, which has been successfully employed on a large scale, is to make the radiating system perform the work of a cooling tower or pond, thus utilizing the heat that is commonly wasted. This arrangement is shown in diagram in Fig. 2 and employs a surface condenser with the usual pumping outfit. The cooling water, as it leaves the condenser, instead of being discharged to the sewer or sent to a cooling tower, is pumped through the radiation in the same manner as in the ordinary system of forced hot-water circulation and, after being thus lowered in temperature, is returned to the condenser and used over again the same as when a cooling tower is employed.

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As the temperature of the discharge water from a high-vacuum condenser is considerably lower than that required in the ordinary heating system in the coldest weather, a compromise must be made, which may be brought about by lowering the vacuum somewhat, and also reheating the water after it leaves the condenser, either by heaters utilizing the exhaust from auxiliaries, supplemented by live steam if necessary, or by economizers placed in the smoke flue, or by a combination, as shown in Fig. 2. The discharge water leaving a need be cooled only to approximately 135 deg. to maintain a ratio of 70 between cooling water and steam condensed. Cooling the return water is accomplished by manipulating valves *A*, *B* and *C*, which discharge a part of the warm return water to the sewer and replace it with cold water from the main. The proper relation of vacuum, quantity of water circulated and warm water wasted, for the most satisfactory results, must be determined by experience in each particular case. As the outside temperature rises, the water may be

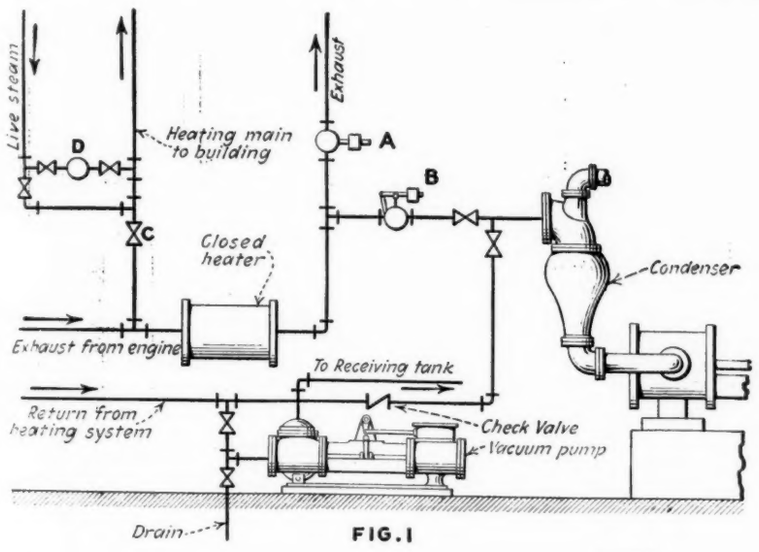

FIG. 1

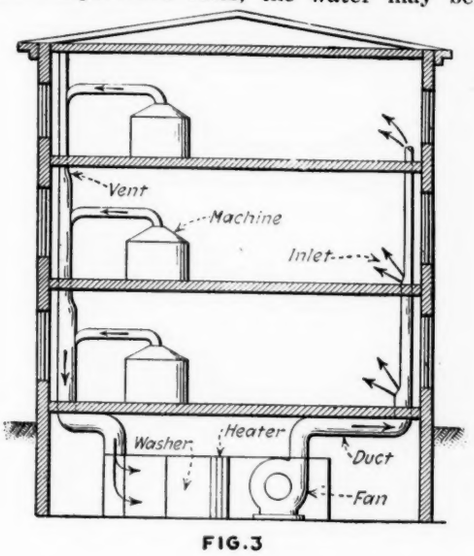

FIG. 3

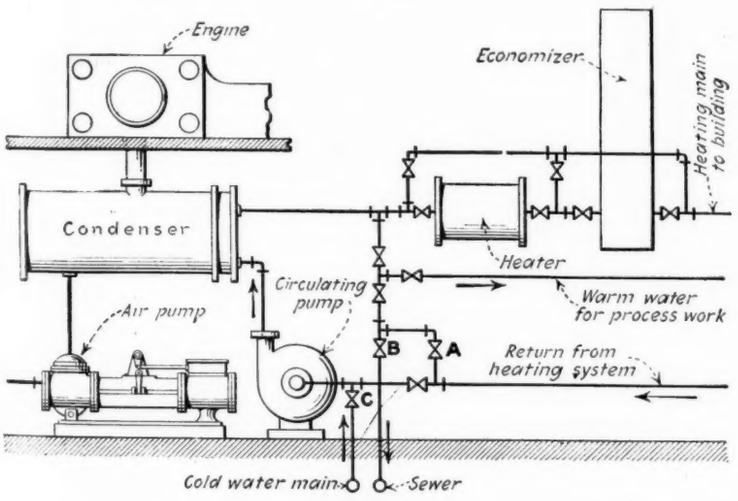

FIG. 2

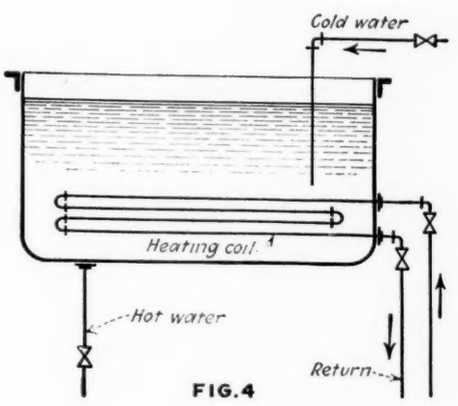

FIG. 4

FIGS. 1 TO 4—ARRANGEMENT OF AUXILIARIES AND PIPING CONNECTIONS FOR RECLAIMING HEAT FROM CIRCULATING WATER AND PROCESS

Fig. 1—Connections for passing exhaust steam direct to heating system or to condenser. Fig. 2—Connections for utilizing condenser circulating water in heating system or for process. Fig. 3—Method of utilizing heat generated by machines in heating building. Fig. 4—Open-type heater and storage tank.

condenser carrying a vacuum of 24 in., will have a temperature of 125 to 130 deg. F. under average conditions, and for radiation of the usual proportion must be raised to 180 or 190 deg. in the coldest weather. Allowing a maximum drop of 40 deg. in the heating system, the water will come back at 140 to 150 deg. and must be cooled to approximately 100 deg. in order to maintain the usual ratio of 35 lb. of cooling water per pound of steam. By pumping 70 lb. of water per pound of steam through the condenser, the return water need be cooled only to 115 deg. If the vacuum is allowed to drop to 20 in., the discharge water will leave at about 150 deg. instead of 130, and the return water circulated at a lower temperature, which calls for less cooling, and the point is soon reached where the return comes back sufficiently cool to maintain a fair vacuum without the addition of cold water and the bypassing of hot water to the sewer.

A certain amount of experimenting will enable the engineer to prepare a chart of corresponding vacuums, water temperatures and pump speeds for the most economical results under varying outside weather conditions. Once this chart is prepared it should enable the engineer to operate the different auxiliaries at the proper speed or capacity to secure the best results under varying conditions.

In checking up temperatures and quantity of water to be circulated, the following formula may be used:

$$W = \frac{H - T_2 + 32}{T_2 - T_1}$$

in which

W = Pounds of cooling water per pound of steam condensed;

H = Total heat in 1 lb. of steam at condenser pressure;

T_1 = Initial temperature of cooling water;

T_2 = Temperature of discharge water.

The accompanying table gives values of H and temperature of steam at different condenser pressures.

VALUES OF H AND TEMPERATURE OF STEAM AT DIFFERENT CONDENSER PRESSURES

Vacuum in Inches of Mercury	Absolute Pressure, Lb. per Sq. In	Temperature, Deg. F.	Total Heat (H)
16	7	177	1,137
18	6	170	1,134
20	5	162	1,131
22	4	153	1,127
24	3	142	1,122
26	2	126	1,115

Under ordinary working conditions W commonly runs from about 35 lb. for the vacuum carried with reciprocating engines up to twice that for turbines where a high vacuum is required. In using the formula for the lower vacuums commonly employed with reciprocating engines, T_2 is taken from 10 to 15 deg. below the temperature corresponding to the condenser pressure. In turbine installations this is reduced to about one-half the figures given.

With the lower vacuum carried, when the cooling water is circulated through a heating system, T_2 should be from 12 to 15 deg. lower than that of the steam at condenser pressure.

Example: In a certain plant it is necessary to carry an initial water temperature of 180 deg. on the heating system in zero weather. Assuming a vacuum of 22 in. at such times and a water circulation equivalent to 60 lb. per pound of steam condensed, how much will the water have to be reheated after leaving the condenser for the heating system, and how much will the return have to be cooled, assuming a drop of 40 deg. in passing through the radiation? Here we have: $W = 60$; condenser pressure (22-in. vacuum) = 4 lb. absolute; temperature of steam at condenser press = 153 deg.; $T_2 = 153 - 12 = 141$ deg.; $H = 1{,}127$; discharge water to be reheated, $180 - 141 = 39$ deg.; temperature of return water = $180 - 40 = 140$ deg. Solving for T_1, the temperature at which the cooling water must enter the condenser, we have $\frac{1{,}127 - 141 + 32}{141 - T_1} = 60$, from which $T_1 = 124$ deg. and the return must be cooled $140 - 124 = 16$ deg. by the addition of cold water from the main.

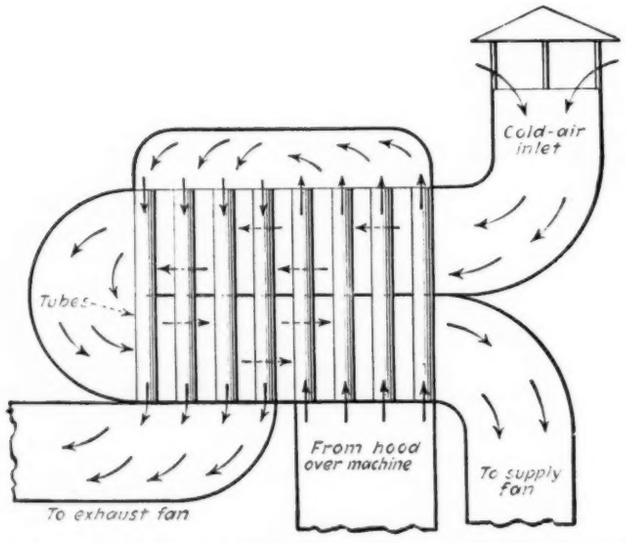

FIG. 5—METHOD OF RECLAIMING HEAT GIVEN OFF BY DRIERS OR OTHER MACHINES

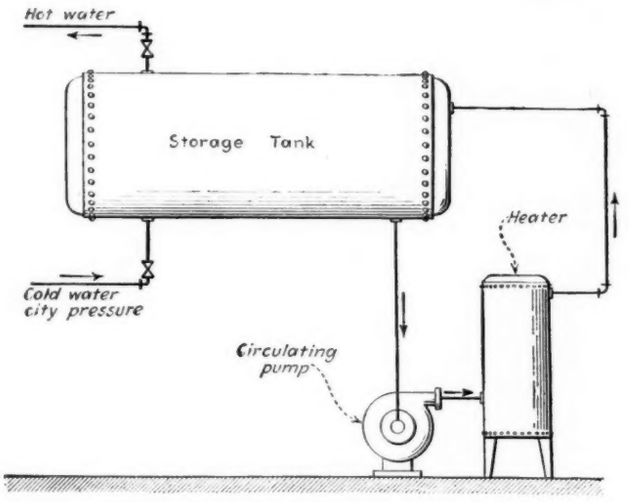

FIG. 6—CLOSED TYPE OF HEATER AND STORAGE TANK WITH FORCED CIRCULATION

Most industrial plants require more or less hot water for process work, and instead of wasting the warm water which must be drawn off at valve *A*, Fig. 2, in the cooling process, it can be diverted for manufacturing purposes or used as makeup in boiler feeding if it is of suitable quality.

In case the weather conditions in the problem cited, were such that a maximum water temperature of only 164 deg. was required in the heating system, the return would come back at $164 - 40 = 124$ deg. and no cooling would be required. This condition would of course hold for all initial water temperatures in the heating system below 164 deg. for the assumed conditions.

The principles involved in both of the arrangements described have been successfully used in practice, the latter in a large plant containing nearly 80 miles of piping in the heating system.

Saving the Frictional Heat

Another method of utilizing waste heat, which has first been put into practice on a large scale, is that of saving the frictional heat generated by certain manufacturing processes and also that from the transmission machinery.

The particular case referred to is a large flour mill where a good deal of heat is generated in the grinding process. It is estimated that the saving made possible by the system employed in this plant will amount to at least 300 tons of coal per year. Other cases should be suggested by this scheme where heated air from various frictional processes, including that from belts and shafting, can be collected, purified and returned to the building instead of being discharged out to the atmosphere as is usually the case.

The general principle involved is shown diagram-

matically in Fig. 3, in which the heated air is drawn from openings near the ceiling or from hoods over the individual machines generating the heat, or from both sources, as the case may be, and carried to the basement. Here it is purified, usually by passing through an air washer, reheated as may be necessary and returned to the building by a fan, as indicated in the drawing. No figures can be given for this class of work, as conditions vary so widely, the only practical way being to collect the heated air in any given case, and reheat it by passing over steam or hot-water coils,

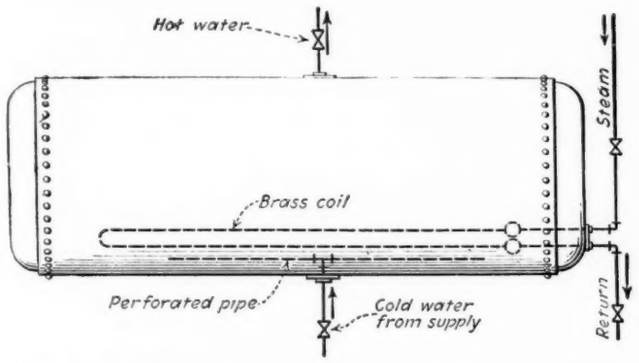

FIG. 7—PRACTICAL AND EFFICIENT TYPE OF HEATER AND STORAGE TANK

to such a temperature as may be necessary to warm the building properly.

Other cases where large quantities of heat are wasted, include paper-machine rooms, dyehouses and other plants where large volumes of vapor are discharged to the atmosphere from the various manufacturing processes.

Ventilating with Dry Air

The most approved method of ventilating buildings of this kind is to introduce a generous amount of warm dry air near the floor and take it off, with the vapor which it absorbs, through openings in the ceiling. In the case of a paper-machine room the air is usually introduced at a temperature of 65 to 70 deg., the actual point being governed by the heat required to maintain a temperature between 70 and 80 deg. in the room. The heat from the rolls is ordinarily sufficient to raise the temperature of the air rising from the driers to 100 or 110 deg. with the room temperatures previously mentioned. Air from the outside at an average winter temperature of 35 deg. and 55 per cent relative humidity, will, when passed through the room and discharged at 105 deg. with a humidity of 97 per cent, carry off 3 lb. of water vapor per 1,000 cubic feet.

If 1.5 tons (3,000 lb.) of vapor is given off in the drying of each ton of finished paper, then $\frac{3,000}{3} \times 1,000 = 1,000,000$ cu.ft. of air must be passed through the room for each ton of paper produced. If this moist air could be cooled to 80 deg., the heat given up would amount to approximately 2,000,000 B.t.u.

Saturated air at 80 deg. temperature contains nearly 1.5 lb. of moisture per 1,000 cu.ft. or half the quantity at 105 deg. and 97 per cent humidity, so that cooling 1,000,000 cu.ft. of the air to 80 deg. would condense 1,500 lb. and cool it 25 deg.

The latent heat of steam at a temperature of 105 deg. is nearly 1,030, so that $1,500 \times 1,030 = 1,545,000$ B.t.u. would be given up in the process of condensation and $1,500 \times 25 = 37,500$ B.t.u. in cooling through 25 deg. The heat given up by the air in cooling through the same range would be $0.018 \times 1,000,000 \times 25 = 450,000$ B.t.u., making a total of $1,545,000 + 37,500 + 450,000 = 2,032,500$ B.t.u. (0.018 being the specific heat of 1 cu.ft. of air at atmospheric pressure).

A boiler operating at 68 per cent efficiency and using coal having a calorific value of 13,500 B.t.u., utilizes about 9,000 B.t.u. per pound, so that if the heat given off by cooling the outgoing moist air from 105 to 80 deg., under the stated conditions could be utilized, it would amount to approximately $2,000,000 \div 9,000 = 223$ lb. of coal per ton of paper produced.

Reclaiming Heat from Ventilating Air

The same method of reasoning of course applies to any other case where large volumes of hot air near the point of saturation are discharged to the atmosphere. Whether it is practical to save any considerable part of this heat in any given case will depend on local conditions, but where such a large amount is involved it is well worth considering.

By blowing this air over coils through which cold water is passing, a considerable amount of heat can be abstracted which may be turned to useful purposes in process work or even in boiler feeding where the exhaust steam is in demand for other uses. As water vapor is practically pure, it might be more effective to heat the water by direct contact; that is, by drawing the air through a spray consisting of the water to be heated. In this case the air must necessarily be free of all impurities in the way of dust particles, etc.

Take again the case of the paper mill. For ventilating the machine room by the method described, 1,000,000 cu.ft. of air must be heated from the outside temperature up to 65 or 70 deg., which will average

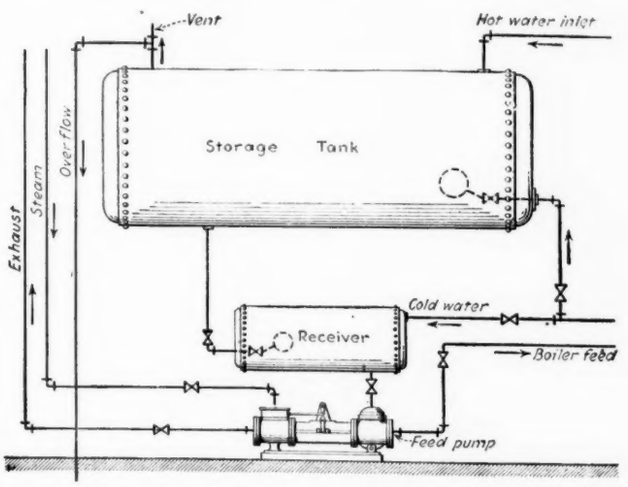

FIG. 8—ARRANGEMENT FOR UTILIZING HOT WATER FROM PROCESS FOR BOILER FEED

a rise of about 35 deg. throughout the heating season, and require $0.018 \times 1,000,000 \times 35 = 640,000$ B.t.u. per ton of paper produced.

Unless there is exhaust steam going to waste, this heat must be drawn from the boilers and represents a given quantity of fuel. By cooling the moist warm air being discharged to the atmosphere 10 deg., or down to 95 deg., more than enough waste heat could be reclaimed to do this work and allow a final temperature

difference of 25 deg. between the two sides of the heating surface, separating the incoming and outgoing air.

While the passage of heat from still air to still air through thin metal is comparatively slow and would require a large amount of surface for a heater of this kind, the conditions in this particular case are much more favorable for a rapid transmission of heat. This is true because of the rapid velocity of the air over the walls of the heater, being in opposite direction on the two sides of the metal, which might easily be raised to an effective velocity of 2,000 + 2,000 = 4,000 ft. per minute.

Again, the condensing moisture will keep one side constantly wet, and as the larger portion of the heat comes from the latent heat, the actual conditions will partake to a considerable extent of the transmission of heat from water to air, which is very rapid. The general principles of such a heater are illustrated in Fig. 5.

Reclaiming Heat from Moisture-Laden Air

In textile mills, where large quantities of hot water are required for washing, etc., and also where large volumes of steam or moisture-laden air must be removed from dyehouses, it would seem that the method of passing the air over pipe coils might be used for partial water heating, say for preliminary warming, up to 70 or 80 deg. Possibly a combination of air heating and water heating might be employed in some cases by inserting the water coils in some part of the heater shown in Fig. 5.

Mention of textile-mill requirements suggests improvements in the usual methods of heating water for process work by means of exhaust steam and storing the same for use when needed. It hardly seems necessary to call attention to the waste involved in providing and operating a condenser to increase the efficiency of the prime mover from 10 to 20 per cent and then drawing on the boilers for live steam for heating water for process. However, this fact seems to be overlooked in some cases, even at the present time of high fuel costs.

By inserting a heater in the exhaust line between the prime mover and condenser, the advantage of condensing may still be retained and large quantities of water heated to 105 to 120 deg., which often answers for many purposes. Higher temperatures may be obtained by supplementary heaters using exhaust from the auxiliary pumps, economizers in the smoke connections or, if necessary, by the use of live steam after waste heat has been utilized as far as possible.

Types of Storage Tanks

Another point where improvement over usual methods may often be made is in the form of storage tank. Two common arrangements are shown in Figs. 4 and 6. The tank shown in Fig. 4 is open to the atmosphere, and the water must be drawn by gravity from the bottom of the tank, while the hottest portion at the top is never reached, and radiates much of its heat to the atmosphere. In the arrangement shown in Fig. 6, although the tank is closed and the water at a uniform temperature, due to constant circulation by the pump, any drawing of hot water will cause an inrush of cold water which is at once mixed with the entire contents of the tank and soon lowers its temperature to a point where live steam must be used to bring it up to the desired temperature.

The simplest and most effective method is that shown in Fig. 7. This has all the advantages and at the same time does away with the auxiliary heater and pump shown in Fig. 6.

This arrangement consists of a closed tank connected with, and under the pressure of, the city mains, reservoir or other source of supply. Cold water is admitted at the bottom, directly beneath the heating coil, and hot water is drawn from the top. The steam coil, which should be of brass or copper, is submerged in the coldest water and therefore capable of the highest efficiency. As the cold water enters through a number of small holes drilled in the sides of the pipe, there is practically no agitation or mixing of the water before it is heated, and the hottest water is always drawn from the top of the tank. The exhaust from the engine or turbine, or such portion as is necessary, passes into the coil at all times, so that water is being heated and stored continuously regardless of whether it is being used or not.

It sometimes happens in manufacturing plants that considerable quantities of hot water are available at irregular periods, as from steam presses or other mechanical processes. This water may often be utilized for boiler feeding or other purposes by means of an arrangement similar to that shown in Fig. 8. A large storage tank is placed above the pump and receiver and provided with an overflow and vent. The hot-water supply enters at the top, and the bottom of the tank is connected with the pump receiver by means of a ball cock which admits water to the receiver as needed. When the hot-water supply is insufficient for boiler requirements, cold water is admitted to the storage tank through a ball-cock located near the bottom.

Other devices along the same line could be mentioned, but those noted will serve as suggestions for investigation and possible improvements in the over-all efficiency of the average plant.

Intelligent Valve Setting

By N. H. Hite

A properly constructed ideal indicator diagram is of great assistance to the engineer when he is setting the valves of a steam engine, steam pump, or air compressor. Taking a series of indicator diagrams from a steam engine whose valves are adjusted after each trial is an unnecessary expenditure of time and energy.

An ideal diagram drawn on tracing cloth to the same scale as the actual diagram, when placed over the latter will show at a glance the defects in valve setting and will also serve to indicate the proper methods to be taken in correcting them. The ideal diagram, once constructed, may be used as a permanent means of comparison for the machine in question.

The expansion of steam in an engine cylinder when shown graphically on the indicator diagram, resembles the equilateral hyperbola more closely than any other curve; that is, the product of the pressure and volume at any point on the expansion stroke is equal to the product of the pressure and volume at any other point on the stroke. For this reason it is customary to construct both the expansion and compression curves of a theoretical diagram as equilateral hyperbolas.

In constructing a diagram of this nature, first ascertain the operating conditions. The necessary data are

the maximum steam pressure in the cylinder, the required cutoff, the pressure at the end of compression, and the back pressure on the piston. The length of the diagram should be the same as the one obtained by the indicator. As an example, if an engine has a stroke of 18 in. and the reducing mechanism of the indicator produces a diagram one-sixth size, the theoretical diagram should be 3 in. in length. Also, if a 60-lb. spring is used in the indicator, all pressures should be drawn to a scale of 1 in. equals 60 lb.

Fig. 1 is such a diagram constructed for, say, an engine having a 36-in. stroke and equipped with an indicator using a 60-lb. spring. The diagram length, with a 1 to 12 ratio should then be 3 in., and the pressures should be drawn to a scale of 60 lb. equal 1 inch.

As the pressures of an ideal diagram are expressed in pounds per square inch absolute, they should be laid off from the line indicated as the vacuum line in Fig. 1. The latter line is laid off below the atmospheric line at a distance proportional to 14.7 lb., as this is the average value of the atmospheric pressure. The pressure of the atmosphere varies slightly, depending upon the height above sea level and the location, but the value 14.7 will prove sufficiently accurate.

The clearance is assumed to be 3¼ per cent of the stroke and is represented in Fig. 1 by two lines drawn at either end of the diagram. The distance between the clearance line and the end of the diagram is then 0.03¼ the length of the stroke. Assuming that cutoff occurs at one-fourth stroke, a line, as *AB* should be laid off parallel to the end of the diagram and at a distance equal to one-fourth the length of the stroke.

The intersection of the clearance line and the vacuum line is taken as the origin of the construction necessary in locating the expansion curve or compression curve. These two points are shown at *O* and *O'*. With *O* as the origin draw any number of lines as *OA*, *OC*, *OE*,

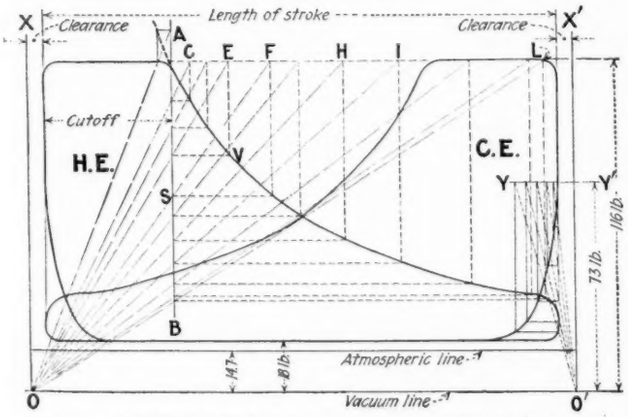

FIG. 1—CONSTRUCTION OF AN IDEAL STEAM ENGINE INDICATOR DIAGRAM

OF, *OH*, *OI*, *OL*, etc. As an example of the method used in locating a point on the expansion curve, a line *OF* is drawn from *O* to any point *F* as shown and this line intersects the line *AB* at *S*. Point *S* is projected horizontally to the right and point *F* vertically downward; the intersection of these two projection lines results in a point on the expansion curve of the head-end diagram. By drawing any number of lines such as *OF*, it is possible to locate as many points on the expansion line as may be desired.

Assuming that the pressure at the end of compression is to be 73 lb. per sq.in. absolute, a line is drawn parallel to the vacuum line and at a distance proportional to 73 lb. drawn to a scale of 60 lb. equal 1 in., as that is the scale used for the remaining pressures. This line is shown as *YY'*.

Point *O'* is used as the origin of the construction employed locating the compression curve of the crank-end diagram. Draw any radial line until it strikes *YY'*.

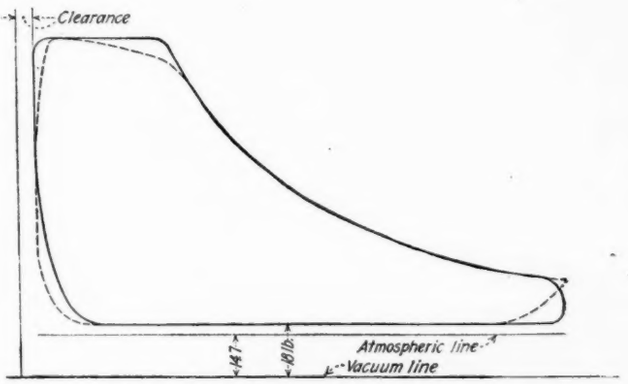

FIG. 2—COMPARISON OF ACTUAL AND IDEAL DIAGRAMS

The latter point of intersection is projected vertically downward and the point of intersection of the radial line and the end of the diagram is projected horizontally to the left, the point of intersection of the projecting lines being a point on the compression curve. Extreme accuracy must be practiced in locating the points of the compression curve by reason of the small scales used on diagrams obtained by the use of an indicator.

All curves should be carefully drawn and the corners of the diagrams rounded off, as it is impossible to obtain a diagram having sharp corners. The most common error made in the construction of diagrams of this nature is the failure to work from the vacuum line.

The diagram so constructed is the ideal at which the engineer should aim. By transferring this to tracing paper or cloth, it may be laid over a diagram taken from the engine and defects discovered as illustrated in Fig. 2. The dotted lines form the actual diagram and the comparison reveals that all events, admission, release and compression, are late. The engine giving the diagram was a single-eccentric Corliss, and it is evident that the eccentric has not enough angle of advance.

A loaded turbine shaft has a natural period of vibration. This corresponds to a certain speed in which an upward and downward vibration occurs at each revolution, which is termed the critical speed. Violent vibration would take place if the machine was run under these conditions. Turbines are designed to operate above or below the critical speed. A "stiff" shaft is made sufficiently rigid to have a much higher critical speed than the normal speed of operation. A flexible-type shaft, however, has a lower critical speed than the normal. The tendency nowadays is to design turbines with stiff shafts, although many builders use flexible shafts for some special conditions.

If there is an abundance of air with CO in the gases, insufficient mixing is the trouble. In most cases there is plenty of oxygen in the furnace, but it goes through in streams and does not mix with the combustible.

Schenectady Type General Electric Valve Gear*

By EUSTIS H. THOMPSON†

THE early turbines built at Schenectady were equipped with an electrically operated valve gear, which was soon abandoned in favor of the hydraulic type. The success that attended the adoption of the latter gear is reflected in the present turbines, which in the main, operate successfully with the same general original features. Improvements relating to operating convenience have in some instances been desirable.

In general, a comparatively slow-speed governor is geared to the turbine shaft. Heavy flyball weights opposing a strong spring in tension, operate a pilot valve or oil relay. This controls the oil supply and discharge from both ends of the operating cylinder, causing the piston to move in proper relation to that of the governor weights. The General Electric Co. has avoided the use of dashpots with this gear, and friction is eliminated or reduced as far as possible at every point where an appreciable bearing pressure exists in order to insure the maximum possible quickness and accuracy in operating.

PRACTICALLY all large and most medium capacity turbines are equipped with some form of hydraulic valve gear. The General Electric Schenectady type is extensively represented. Friction of the controlling parts is reduced to the lowest practical amount, and no dashpots are used. Heavy flyballs and spring supply energy for quick, accurate speed adjustment. The synchronizing spring increases speed by reducing the force against the governor lever. This tends to decrease pressure on the transmission bearing at the heavy loads requiring maximum speed adjustment. Other articles in this series treat of governors and valve gears in a more general manner.

The hydraulic cylinder requiring oil at 60 to 100 lb. per sq.in. supplies ample power through a rack and pinion for opening and closing a number of valves at the top of the turbine. These valves are of the unbalanced type, where the steam pressure assists in seating them. The steam pressure unbalance must be overcome when the valve is lifted. Each of the valves communicates with a group of nozzles in the interior of the turbine. Steam is admitted by opening the valves in succession, so that throttling is accomplished by only one valve at a time, as all others are wide open, admitting steam at full pressure, or else tightly closed. This makes for maximum steam economy by reducing the losses due to throttling to a degree not obtained when only one or two inlet valves are used.

Fig. 1 illustrates the general construction of this gear as applied to a 7,500-kw. turbine. The actual design of the pilot valve, oil cylinder, steam chest, etc., varies to some extent in units of different capacity. In Fig. 1 the oil is supplied by a rotary gear-type pump at the lower end of the governor shaft, and the pressure is adjusted by the relief valve at *C* to 60 or 100 lb. On most large machines an auxiliary steam pump or turbine-driven pump is arranged for supplying oil when the turbine is shut down or being started up. The pipe at *A* which enters the housing or casing around the governor, otherwise known as the governor dome, is for the purpose of sending an oil spray into the moving parts for complete lubrication. Pressure gages for the hydraulic and lubricating systems are mounted on the gear case cover. The details of the design in Fig. 1 differ somewhat from those in Figs. 2 to 7.

If the turbine should be brought up to speed, the governor weights at Fig. 5 would tend to fly apart. The general effect would be to lower the governor stem and therefore the governor lever. It will be seen in Fig. 2 that the lowering of the governor lever will tend to lower the pilot-valve stem on account of the connecting links. Oil will then circulate in the direction of the arrows, Fig. 7, supplying a pressure on the bottom of the oil cylinder and opening the upper part to discharge, causing the piston to rise and rotate the pinion, as indicated by the arrow. This would close the valve, as shown in Fig. 4, the dotted lines indicating the wide-open position. The cams along the shaft, Fig. 3, are set at different angles so that the overload valve to the right is closed first, then the next begins to close, and finally the valve at the extreme left operates only at no load.

The compensating action takes place from the effect of the oil cylinder movement. This raises the link at *C*, Fig. 7, and therefore tends to bring the pilot valve up to a neutral position, where it will prevent movement of the oil contained in the top and bottom of the cylinder. A certain travel, say one-sixteenth of an inch on the governor stem, will therefore produce a corresponding travel of perhaps one inch on the piston of the hydraulic cylinder before the compensating, or floating, lever closes the pilot valve. If load now be put on the unit, the opposite sequence of events takes place with the compensating lever checking the movement of the pilot valve in direct relation to the travel of the governor spindle.

Tension-Spring Main Governor

The synchronizing spring, which opposes the governor spring, is used to increase or decrease the speed a small amount, usually 4 per cent above or below normal. This is for the purpose of properly handling the load of an alternating-current generator. It is operated by an electric motor which drives by a worm gear the adjusting shaft of the synchronizing spring. In order to prevent the motor being operated too far in either direction, automatic cutout switches are provided, which open the motor circuit when the limit of the synchronizing spring travel has been reached and therefore shut off the motor power in that direction. The motor may be reversed, however, until the opposite limit switch is opened.

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The governor shown in Fig. 5 utilizes a spring in tension, operated by heavy flyball weights below. The main governor lever is pivoted to the right with roller bearings so that its weight acting downward through the governor spindle is supported by the main governor spring. The synchronizing spring is in compression, which also adds force downward in opposition to the governor spring. This makes appreciable tension. Somewhat less movement is therefore necessary in screwing up the adjusting nut at the top of the governor than would be required if this nut supplied all the tension.

The main governor stem contains a universal joint at the lower plug. This furnishes flexible connection, so that in case the lower plug does not revolve in the same axis as the governor driving shaft, the governor stem will still be able to align itself properly with the gimbel bearing on the governor lever.

The governor dome is stationary, as indicated in Fig. 5, while the revolving governor base supports the structure shown at the top from which the main spring is suspended.

The main governor flyball weights pivot on the hardened steel knife-edge blocks at the bottom of the upper cage shown in Fig. 5 and are connected to the lower plug by links containing knife-edges. They are shown in the slow-speed position which would allow all the main valves to be open. When the machine speeds up, these weights fly apart. In case of overspeeding they are prevented from reaching an abnormal position by means of the vertical stops, *F* inserted in the revolving base. The weights are grooved so as to clear the plugs until the time the entire range of opening is reached.

Speed is raised by tightening the adjustment nut at the top for increasing the spring tension and therefore the no-load speed of the turbine. The upper plug is prevented from turning by means of the key shown at the right, which is attached to a plate lying between the adjusting nut and the upper governor cage. This plate is prevented from turning by means of the screw *G* shown at the left, tying it to the cage.

The governor stem is prevented from turning by the spring which forms part of the flexible joint. This is locked by the nut at the top. The spring cannot change its position at the bottom, as any friction would tend to screw it up on this plug to the limit of the thread and therefore jam so as to prevent further turning of the spring.

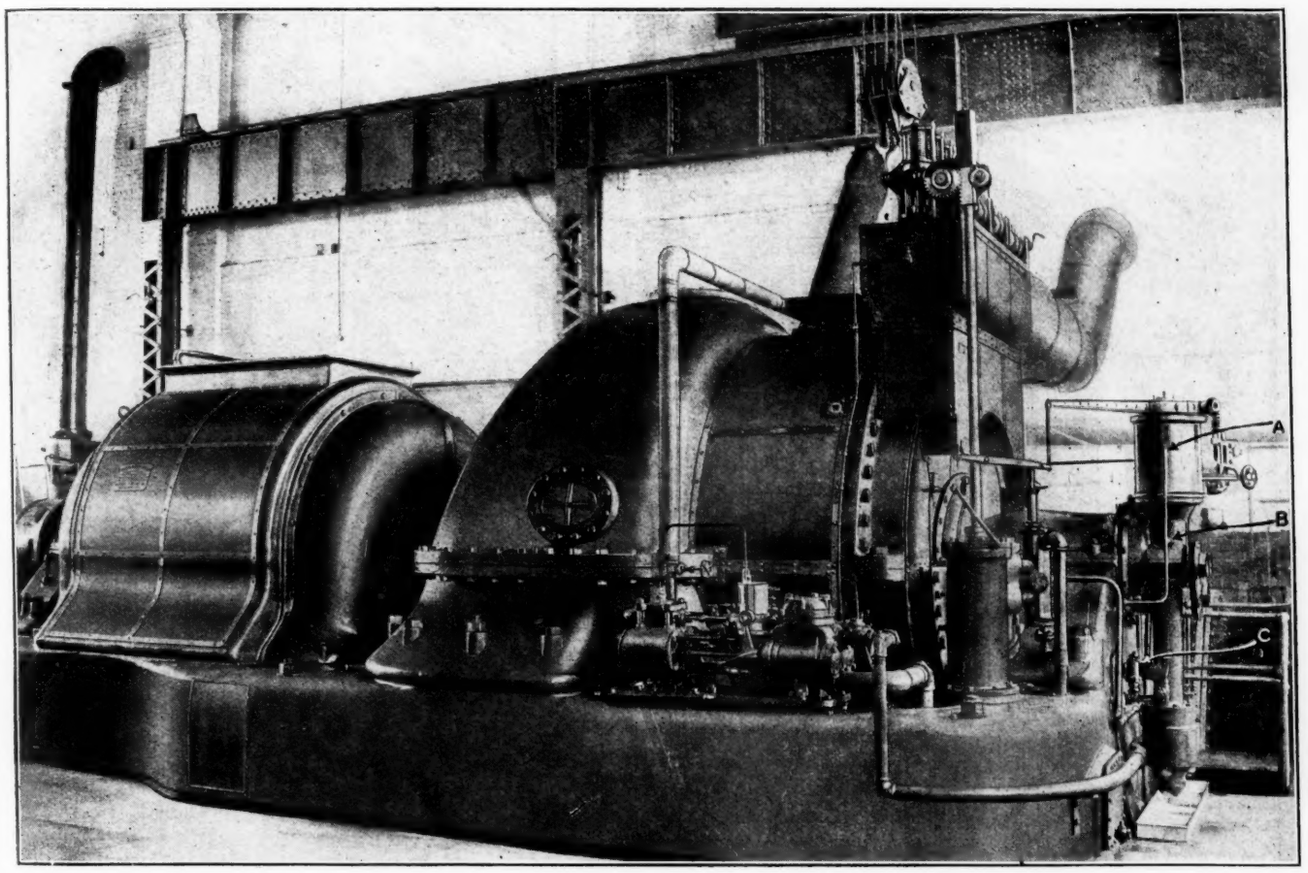
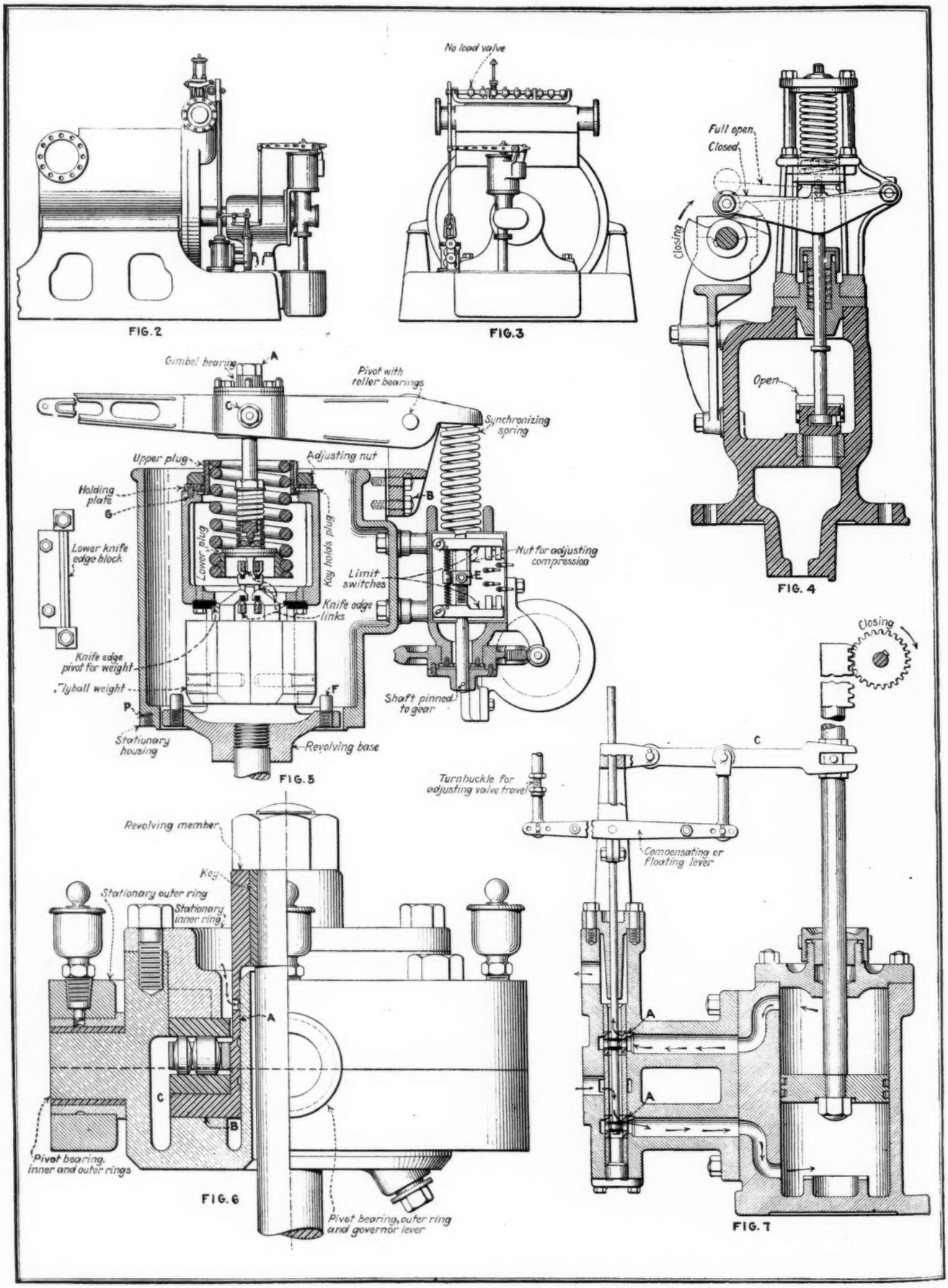


FIG. 1—HYDRAULIC VALVE GEAR ON A GENERAL ELECTRIC 7,500-KW. UNIT

Governor Regulation Can Be Adjusted By Means of Changing Weight or Active Length of the Governor Spring

The synchronizing motor operates a large gear which is pinned to the mainshaft. This shaft is threaded into a nut at the top which changes the compression on the synchronizing spring according to the rotation of the motor. This nut also carries an engaging member *E* which will open one of the motor current limit switches when the limit of the spring adjustment is reached, shown more clearly in Fig. 9. The gear operates in a case containing light lubricating grease. Balancing holes are usually provided in the upper cage for adjusting the balance of this governor. In another design of



FIGS. 2 AND 3—GENERAL ELECTRIC VALVE GEAR ELEMENTS. FIG. 4—CAM RAISES AND SPRING CLOSES STEAM VALVE. FIG. 5—COMPRESSION-TYPE SYNCHRONIZING SPRING CONTROLS TENSION SPRING GOVERNOR. FIG. 6—GIMBEL BEARING ALIGNS ITSELF WITH SPINDLE. FIG. 7—PILOT VALVE REGULATES HYDRAULIC PISTON

this governor, Fig. 8, the construction permits the operating parts to be lifted out of the cage entirely.

In this governor there are two methods of changing the regulation or in other words, of adjusting the number of revolutions that are required between no-load and full-load valve openings. A turbine operating

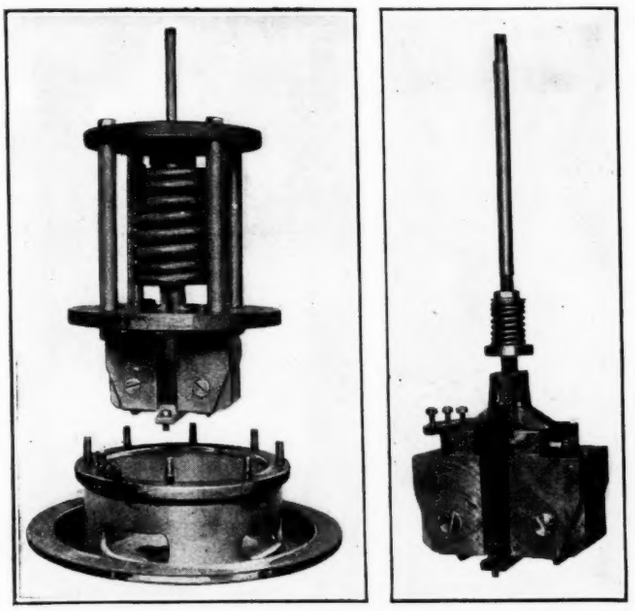


FIG. 8—GOVERNOR PARTS EASILY REMOVED

at 1,800 r.p.m. with a 2 per cent regulation, would ordinarily open the valve sufficiently to carry full load at 1,764 r.p.m., which is 36 r.p.m. below no-load speed. If the no-load speed is correct, yet the full-load speed is 1,750, the governor regulation of 50 r.p.m. is then too broad or, in other words, too many revolutions are required between no-load and full-load opening of the valves.

The regulation can be shortened by adding to the weight of the flyballs or moving governor weights, as they are called, which is the usual method of adjusting regulation. Another method is to increase the number of active turns in the main governor spring, by unscrewing it a small amount off of one or both of the governor plugs. The latter method should be avoided as far as possible. The flyball weights have holes drilled near the bottom portion, as shown in Fig. 5, for inserting plugs. Their weight can be increased by adding plugs in these holes or can be diminished by decreasing the plugs. Each adjustment at this point either lowers or raises the no-load speed of the turbine, so after every adjustment is made, either at the weights or the governor spring itself, the no-load speed must be adjusted accurately before the regulation is checked.

Governor Is Easily Dismantled for Adjustment or Repairs

If it should become necessary to change the main spring by either unscrewing it on one plug or screwing it up more, it should be remembered that a very slight difference in the length will make a comparatively large difference in governor regulation. Therefore, about one-eighth to one-tenth of a turn in the spring would be sufficient for checking its effect on regulation under ordinary circumstances. It is customary to place the synchronizing spring in the mid-position when checking either speed or governor regulation.

In dismantling the governor, Fig. 5, either the governor lever at the top may be lifted off separately or else the entire governor dome, together with the lever, may be removed. Slack the compression in the synchronizing spring as much as possible before doing anything else, by running the motor until the lower limit switch is opened, as in Fig. 9. The first operation requires the link connecting the lever to the pilot valve to be disconnected, and the nut *A* at the top of the stem removed and then holding the nuts *B*, Figs. 5 and 9 on the lever brackets taken off. The lever and upper brackets can now be removed together with the gimbel bearings.

In removing the governor dome and lever together, the governor lever should be disconnected from the pilot valve, and then the pivot bolts at *C*, Fig. 5, removed. The bolts holding the governor dome in place at the bottom, indicated at Fig. 5, should be partly taken out, and also the wiring disconnected from the synchronizing motor. The crane or other means of lifting the dome should now be rigged up with slings taking a little strain and the remaining bolts holding the dome in place removed. The dome and lever can be lifted off together, leaving the gimbel bearing in place on the spindle. It is sometimes more convenient to lift the gimbel bearing off together with the lever by first removing the nuts at *A* instead of those at *C*. This, however, requires somewhat more skill and should be attempted only by thoroughly experienced men.

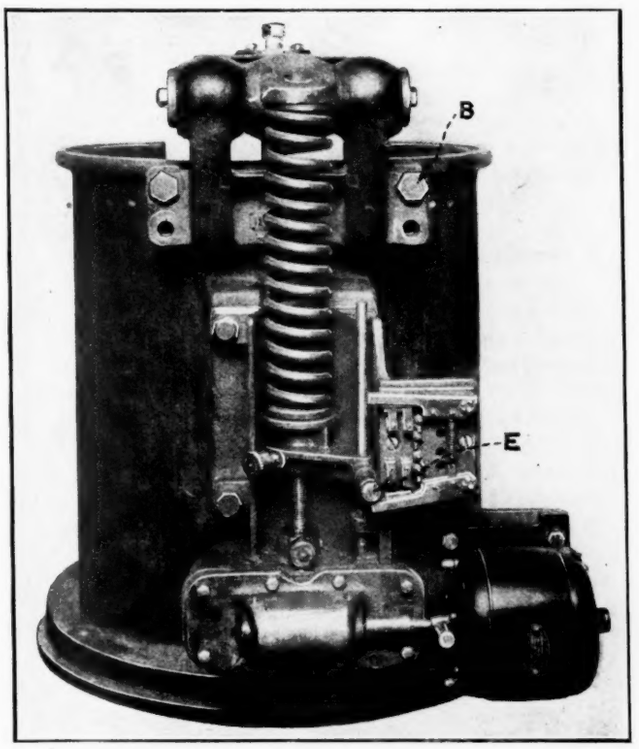


FIG. 9—LIMIT SWITCH STOPS MOTOR IN HIGH-SPEED POSITION

In renewing governor parts, it is customary to loosen the adjusting nut at the top of the upper plug until the spring tension is entirely removed. The governor weights will thus be dropped to the bottom and then disconnected from the links and removed by hand. The

screw connecting the holding plate into the governor dome at *G*, Fig. 5, should now be taken out and the entire spring can be lifted off. The governor, Fig. 8, allows all moving parts to be taken out together. It will be seen that the steel pieces in the knife-edge links, which are shown in full black section, Fig. 5, are separate from the body of the link itself, shown in cross-section, and hence can be readily renewed. The steel pieces and knife-edges at all points are arranged for convenience in removal. When the governor is to be assembled, great care should be used to see that all parts line up correctly. The bolts required for holding the lower knife-edge blocks in place are shown to the left in Fig. 5.

Adjusting Hydraulic Gear

Oil under pressure is admitted to the middle section of the pilot valve between the two spool-shaped valves, as indicated in Fig. 7. As long as the pilot valve remains in the neutral position, oil will be blocked from flowing to the top or bottom of the hydraulic piston. The discharge port at the bottom and that at the top, together with the overflow indicated at the top of this valve, are connected, forming one passage. Steel bushings shown at *A*, Fig. 7, act as pilot-valve seats. Beside these there are one middle and two end guide bushings. The pilot valve itself is hardened steel.

The compensating or floating lever which is connected to the governor lever, may be adjusted in its proper relation to the governor lever by means of the turnbuckle shown in Fig. 7. In adjusting the turnbuckle for the proper valve travel, it is customary to bring the turbine nearly up to speed but somewhat below it, so that the governor has not yet begun to operate, and the weights are closed, as illustrated in Fig. 5. The turnbuckle is then adjusted so that the no-load valve, which should be the first of the valves to close, is just on the point of closing and any small upward travel of the hydraulic cylinder will cause it to begin its closing stroke.

The most delicate adjustment of any hydraulic valve gear is that of the pilot valve. It is customary to place oil pressure on the gear and disconnect the pilot valve from the governor lever, so that the compensating lever may be operated by hand. It can then be felt on moving by hand if any appreciable friction or binding exists. If friction or binding should be found, it should be determined whether this is in the compensating lever and its links or the bearing at the top of the pilot-valve stem, or whether this is due to the friction between the spools and the bushings of the pilot valve. This can be done by disconnecting the various links, one at a time, and trying the effect of moving by hand with and without other links.

If the friction is due to the pilot-valve stem, it should be decided whether this is due to a sprung stem or simply a tight fit between the spools and the bushings. In the former case the stem must be straightened satisfactorily or else a new one provided. In the latter case it is customary to relieve the friction by using a piece of finest-grade emery cloth on the spools, which are revolved in a lathe.

It is quite necessary to take only a small amount, about 0.0005 in., off at a time and make sure whether the friction is in one or both spools, so that a good fit may be produced.

When the pilot valve becomes worn excessively, the discharge oil increases abnormally in volume or a thrust may develop in the pilot-valve stem or a sluggishness of the hydraulic cylinder may be noted. In order to renew the seats, disconnect the floating lever, unscrew the guide bracket bolts and withdraw the valve stem, pistons, sleeve and guide bracket. The bushing should be removed by carefully machining a piece of brass to fit

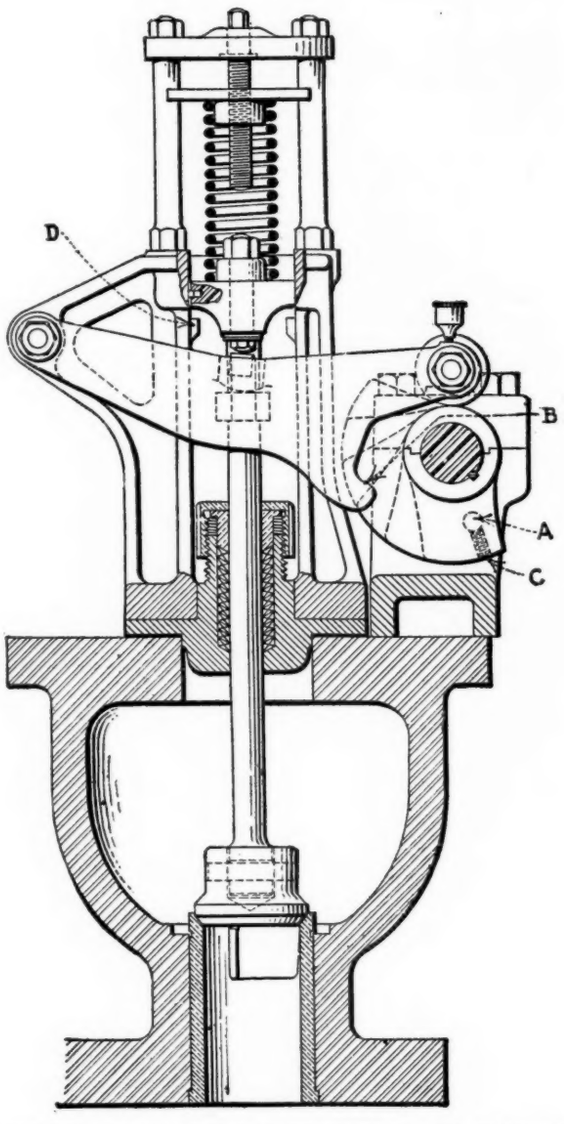


FIG. 10—PIN IN THE CAM INSURES VALVE CLOSURE

the inside loosely and yet contain a shoulder which may be used as the means of applying the driving pressure to the bushing. Line up the interior three bushings by driving a soft-wood stick into them, so they may all be driven out together. This will prevent one of the smaller bushings from being jammed in the aperture that connects to the valve port. Drive out one end bushing, then reverse the direction of driving and remove the bushing at the other end. These are fitted tighter than the others, and it would entail a waste of effort to drive either one the full length of the valve.

Unbalanced Steam Controlling Valves

The steam valve in Fig. 4 is shown in the closed position, and it will be noticed that the steam pressure tends to hold the valve closed. This is raised by the action of the hydraulic cam and is closed by the pressure of the spring at the top. These are of the simplest possible

construction, and there are no delicate adjustments to make. Wear of operating parts and leakage due to the cutting of the valves are about the only difficulties ordinarily experienced, being common to any type of steam valve. Owing to the great seating pressure this leakage trouble is notably less than that found in a similar balanced valve. Steel valve elements may be readily removed for renewal, and the bushing that forms the seat of the valve is also renewable. The cams of the different valves are cut off as indicated by the dotted line in the cam of Fig. 4, so that they will operate the valves successively and not all at one time.

Details of control valve, Fig. 10, are somewhat different. In addition to the usual spring-pressure closing device, there is an emergency closing pin *A* which insures lowering of the valve in case abnormal conditions should tend to prevent the valve from closing. The pin at *A* would in that case contact with the cam at *B*, forcing the valve downward. The setscrew *C* holds pin *A* in place.

Gimbel Bearing Supports Governor Lever

The operation of the universal roller transmission bearing connecting the governor lever to the valve stem, as shown in Fig. 6, can be better understood when it is remembered that the governor lever bears downward through the pivot bearings on the outer ring. The internal member, therefore, supports the roller bearings which operate over a steel washer, while above the roller bearing is another steel washer, which acts in a downward direction owing to the weights and thrusts of the governor lever. The inner stationary ring, therefore, tends to clear the bottom of the revolving member at *B*. The lubricated vertical joint *A*, Fig. 6, where the inner ring and revolving member are brought together parallel to the surface of the spindle, need not be a tight fit. There is not necessarily any appreciable strain carried on these latter bearing surfaces, and therefore no appreciable friction exists.

The inner ring consists of two pieces, bolted together, containing a pivot bearing on each side, consisting of a lubricated steel sleeve between brasses. The outer stationary ring also consists of two halves, with similar pivot bearings for connection to the governor lever. The main thrust bearing consists of thin and wide rollers that are staggered, so that a narrow roller is shown on the outside in Fig. 6 and next to this is a wide roller. Following these two rollers are also a wide and a narrow one, but in this case the wide roller would be at the outside and the narrow one at the inside, representing a staggered arrangement. This tends to prevent grooving of the steel members by the rolling action.

A large oil reservoir *C* provides a bath so that the roller bearings and surface *A* are constantly lubricated. Any overflow through the upper holes lubricates the surface *A*, at the top, before escaping between the governor spindle and the portion of the inner ring. A fairly close clearance in the roller bearing of three to five thousandths of an inch is desirable. On account of the downward pressure of the governor lever this clearance is not as important as would be the case if the external pressure were to vary in both directions, therefore taking up end play first one way and then in the other. In operation this clearance tends to open up the joint *B*.

This general type of hydraulic valve gear is varied to suit conditions in various designs of turbines manufactured by the General Electric Co. Smaller machines with inclosed balanced steam valves, often contain one single pilot-valve bushing instead of the combined steel bushings shown in Fig. 7. The number of valves, link connections, design of hydraulic cylinder, details of governor weights, etc., are arranged to suit the different types of turbines and conditions encountered.

The entire gear should be tested at intervals to make sure there is no sticking or binding in the hydraulic elements or excessive leakage of oil through the pilot valve and that the controlling valves work smoothly and evenly. Disconnect at the turnbuckle and move the hydraulic element by hand, to see that the pilot valve moves without binding and the general action is free and smooth. Observe the valve elements to make sure that all the rollers set in the lifting levers roll freely on the cams, for if they become stuck they will tend to wear flat. See that the valve stems work freely in the packing and are not stuck owing to old packing becoming hard or the valve stems becoming pitted.

Make sure that dirt does not collect on the lugs shown at *D*, Fig. 10, and thus prevent the valve from closing, which would allow the turbine to overspeed at light load. Since the steam quickly deteriorates packing containing rubber, causing the valve stems to stick, be sure to use only graphite-lubricated packing that is free from rubber composition of any kind.

The final adjustment should require the turnbuckle to be lengthened or shortened until the overload valve is just on the point of closing. Other links should be so connected that the floating lever will be in a horizontal position when the valve element is half closed.

The transmission bearing at the top of the governor should be kept filled with oil and prevented from allowing the oil passages to become clogged with dirt. Heating of the gimbel bearing may be due to either lack of oil or closed oil passages. This should not heat or throw oil if in proper condition.

The no-load speed of the governor should be adjusted by moving the adjusting nut at the top of the governor while the synchronizing spring is in the middle position. Regulation should be adjusted by changing the governor weights in preference to changing the number of active turns in the main governor spring. Be sure that the oil for lubricating the interior of the governor is flowing freely. If the gimbel bearing wabbles excessively, it may be corrected to a large extent by turning the governor spring around. This means loosening the nut at the top and changing the holding plate to another position, where it can be fastened again by means of extra holes for the holding screw *G*, Fig. 5, which are provided. Do not turn either plug into or out of the spring.

Hunting may be caused by too close a regulation, binding of parts in the governor, binding of parts of the controlling valves, or of rods and levers in the hydraulic mechanism. Sticking of the pilot valve is usually to blame. This may be due to dirt in the passages or a sprung valve or binding; low hydraulic oil pressure is another possible cause.

An excellent and inexpensive refractory furnace, crucible or drum lining may be made by mixing one part of firebrick dust (crushed firebrick) one-half part of fireclay and one-half part of portland cement. This material, when mixed to the proper consistence with water, may be laid up with trowel and in forms. When dried and put into service, it forms a uniform, tough, lasting lining.

Types of Motor Protection

Location of Protective Devices—National Electrical Code Revisions—Undervoltage Protection—Time-Element Features—Phase-Failure Protection—Shunt-Field Relays—Limit Switches—Overspeed Trips and Interlocks

BY GORDON FOX
Electrical Engineer, Freyn Brassert & Company, Chicago, Ill.

A MOTOR that is not overloaded may be damaged by burning or overheating resulting from insulation failures or accidental contacts that short-circuit portions of the windings. If these faults are such as to cause excessive current intake from the lines, the overload protection serves also as a means to limit the damage resulting from the fault. Some faults, such as a short-circuited armature coil, may not cause the protective devices to operate, as excessive currents are local to the faulty coil and do not pass through the protective devices.

If a motor is operated on a circuit that is either normally or accidentally grounded, a ground in the motor or controller may cause a short-circuit or some important portion of the circuit may be "shunted" or bridged around by the ground. The protective devices may serve to afford a measure of protection against grounds, their efficiency depending largely upon their character and their position in the circuit.

LOCATION OF PROTECTIVE DEVICES

In order to protect a direct-current motor fully, it is desirable to install fuses or other circuit-opening devices in both positive and negative lines. If circuit breakers are used, they may best be made double-pole.

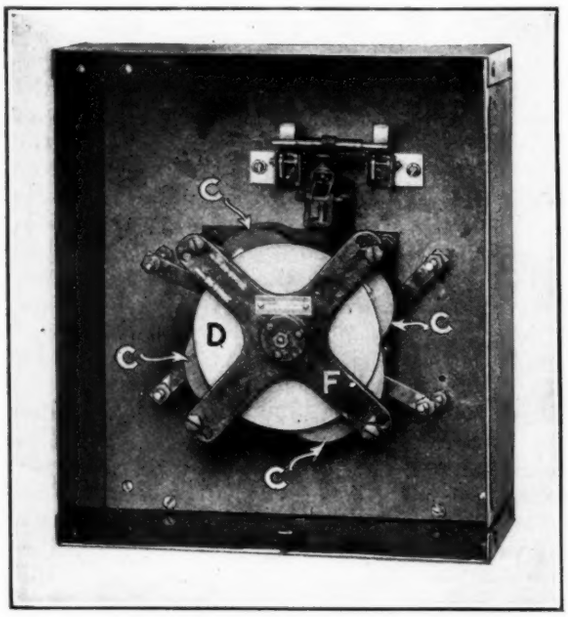


FIG. 1—PHASE-FAILURE AND PHASE-REVERSAL RELAY HAVING FOUR COILS, WHICH ACT ON A METAL DISK

If two trip coils and two contactors are installed, one on each side of the line, the protection is quite complete. Where contactors are employed as circuit breakers, two overload relays or one double-coil overload relay may well be used. Sometimes one relay is of the instant-trip type giving short-circuit protection, while the other has a time element to afford overload protection. This is a good arrangement.

A ruling of the National Board of Fire Underwriters, the justification of which is the subject of some discussion, requires that either fuses or a circuit breaker be installed even where protection is afforded by overload relays and contactors. To meet this requirement, fuses are generally installed ahead of the contactors, the arrangement being such that the fuses blow only in case the regular protection fails. Rule 8c of the National Electrical Code has recently been amended to read:

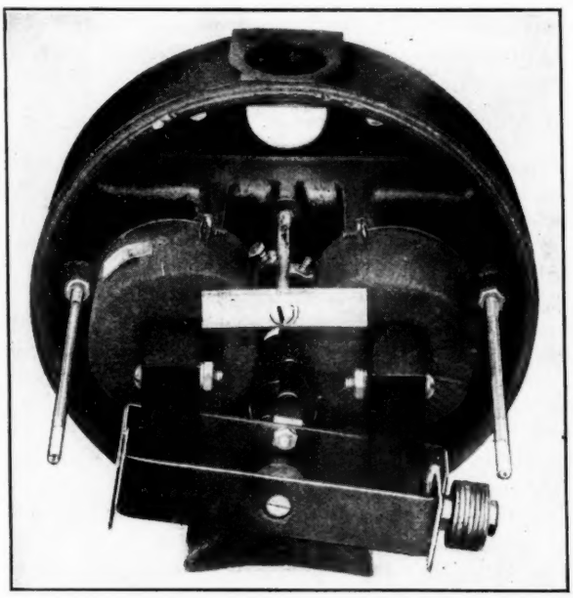


FIG. 2—COMBINATION PHASE-FAILURE AND PHASE-REVERSAL RELAY THAT OPERATES ON THE PRINCIPLE OF AN INDUCTION-TYPE WATTMETER

> Motors may be grouped under the protection of a single set of fuses provided the rated capacity of the fuses does not exceed 15 amperes and the total wattage of the circuit does not exceed 1,200, or provided each motor is protected by thermal cutouts. The number and size of the motors grouped with thermal cutout protection need be limited only by the maximum size of the fuses with which the thermal cutouts can be safely used, and each thermal cutout must be marked to indicate the size of this fuse.
>
> Fuses are not required in addition to circuit breakers (a) on main switchboards, (b) where otherwise subject to competent supervision, or (c) where next back on the line there are fuses rated or a circuit breaker set at not over 300 per cent of the motor nameplate rating, (d) for circuits above the capacity for which there are no standard fuses.
>
> Where the motor-running protective device is shunted during the starting period, the motor and the portions of the motor branch circuit between the motor and its running protective device are considered as being sufficiently protected during the starting period by the next overload protective device back on the line if the rating of this fuse

or the setting of this circuit breaker is not over 300 per cent of the motor nameplate current rating.

Conductors carrying the current of only one motor must have a carrying capacity of at least 110 per cent of the nameplate current rating of the motor, the actual size to be determined by the rating of the fuses or the setting of the circuit breaker protecting them.

Automatic overload protective devices may be omitted at the point where conductors carrying the current of only one motor are connected to the mains, provided their current-carrying capacity is at least one-third that of the mains, the length of the conductors between the mains and the motor protective devices is not greater than 15 ft. and they are suitably protected from mechanical injury.

Undervoltage Protection

Large motors of any type should not be started from rest by connecting them directly across the power lines, it being necessary to modify the electrical conditions during the accelerating period. When a motor is running, its controller is normally on the on position. If the power is interrupted and again established before the controller is returned to its off position, damage may result to the motor, controller or machine. To prevent this occurrence, most controllers are provided with a no-voltage release feature which causes them to return to their off positions when the circuit is interrupted or when the voltage falls to a low value.

If an automatic controller is equipped with a no-voltage release feature, it will return to the off position upon interruption of power. If the master switch remains in the on position, the controller will again start the motor upon return of power, the starting operation being normal. This is a desirable feature for many installations such as fans and pumps, but it is undesirable in cases where a workman might be working on a machine. To prevent accidents from this cause, many automatic controllers are equipped with a no-voltage protection feature. This is similar in action to the no-voltage release except that the controller will not automatically start the motor upon return of power after an interruption, it being necessary for the operator to manipulate his controller to accomplish this.

It is sometimes undesirable that a motor be shut down owing to low voltage, unless entirely necessary. In order to prevent operation of undervoltage devices due to momentary drop of voltage, a time-delay feature is occasionally included.

Phase-Failure Protection

In the case of polyphase alternating-current motors it is desirable to protect against single-phase operation. Such a condition may readily arise, due to blowing of one fuse or faulty contact in one line of a controller. A polyphase motor will not start single-phase, but will continue to run if already under way. Its capacity is materially reduced, and it will generally overheat if called upon to deliver its full mechanical output. The current taken from the remaining phases is increased, and a measure of protection is afforded by the overload devices in these circuits, but if these are not set sufficiently low, overheating is liable to occur. Many motor burnouts are due to single-phase operation.

It is difficult to protect against single-phase operation through use of voltage relays, as the voltage is maintained across the open phase owing to induction-generator action of the motor. The use of undervoltage relays in two phases will protect in a measure against attempting to start with one phase open. Starting devices having a double-throw feature may permit a motor to start polyphase, but the running position may have a faulty contact or a blown fuse, causing the motor to run single-phase. Voltage relays do not protect against this contingency.

The need of a phase-failure protective device that will protect against continued single-phase operation of a running motor has caused the recent introduction of relays to fulfill this requirement. One device of this type operates on the induction principle, in which a revolving field causes a disk *D*, Fig. 1, to rotate through an angle in much the same manner that an integrating

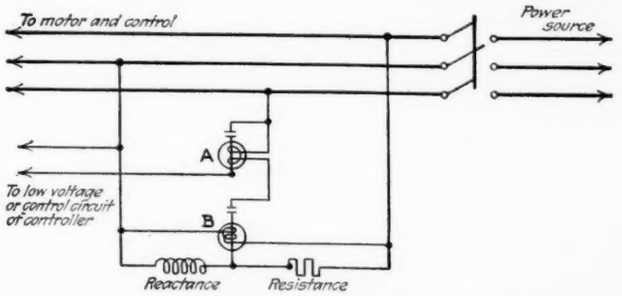


FIG. 3—DIAGRAM OF PHASE-REVERSAL RELAY HAVING TWO VOLTAGE COILS

wattmeter disk is rotated. Coils *C* are connected in series with the motor, directly or through current transformers, are wound on cores provided with shading coils so that, individually, they set up a counterclockwise torque which tends to rotate the disk and thus to open the contacts in a pilot circuit. Acting in combination, when polyphase power is flowing, these coils set up a revolving field which overpowers the single-phase action and causes the disk to rotate in a clockwise direction and close its contacts. When power is cut off all phases, the contacts remain closed, but if one phase only of the motor opens, the polyphase action is lost and the single-phase action causes the relay to open its contacts and thus cause the motor to be disconnected. This single-phase action is not effective on very light loads, but is effective on loads sufficient to cause possible injury to the motor.

In some cases, as in elevator and hoist service, it is advisable to protect against the possibility of a reversal of primary phase relations which would cause reversal of rotation of the motor. Relays that afford protection against this contingency are available. Some of these operate on a wattmeter or induction principle, wherein a torque rotates a movable element to make and maintain a contact. Reversal of phase reverses the torque and opens the contact. Fig. 2 shows a relay of this type. The phase-failure relay, Fig. 1, is of this general description and is also effective as a phase-rotation relay.

Other devices combine two voltage relays that function on the basis of different voltages resulting from different phase relations. The connections for a device of the latter type are shown in Fig. 3. Relay *B* is connected across one phase. The coil of relay *A* is connected to one phase lead and to the junction of a resistance and a reactance. When phase relations are correct, the two component currents through relay coil *A* are cumulative and the relay closes. When a phase is reversed, the component currents are opposed and the relay remains open.

If the shunt-field circuit of a direct-current motor becomes de-energized, the motor loses most of its countervoltage and tends to increase its speed greatly. The

armature current ordinarily increases to a point where the overload relays will trip, but if they are of the time-element type, the action is delayed and the motor may reach a damaging speed. To safeguard against this contingency, a relay is sometimes inserted in series with the shunt field, so that in case of an open field circuit the relay will open and cause the circuit breaker or line contactors to disconnect the motor. This type of relay is particularly applicable to adjustable-speed motors, because of the likelihood of open circuits in the field rheostat. It is somewhat difficult to obtain a relay that will be effective over a wide range of field current values such as may exist with an adjustable-speed motor. Moreover, transformer action between armature and fields may cause momentary drop in field current due to fluctuations in armature load. Damping devices may be necessary to overcome this difficulty.

When Limit Switches Are Provided

In the case of many driven machines a travel occurs, the permissible travel being limited. It is not always safe or may not be desirable to expect the operator always to stop the machine within the safe limits of travel. In such cases it is customary to provide a "limit switch," which is a means for automatically stopping the motor before the limits of travel are exceeded. Limit switches may interrupt or change connections in the motor circuit handling the load current. More commonly they are arranged to make or break contacts in the auxiliary circuits of the controller to govern the action of the controller and cause it to stop the motor.

Ordinarily, limit switches merely cause the motor circuit to be interrupted so that the motor will stop. In some cases they cause a dynamic braking circuit to be made, quickly bringing the motor and machine to rest. In other cases they are arranged not to stop the motor but automatically to change the control circuits or conditions to bring about a change in speed.

Overspeed Trips and Interlocks

In some cases it is possible for a motor to drive a machine at a speed exceeding the safe or the desired rate. In such cases a speed-limit switch is sometimes used, this comprising some type of flyball device arranged to cause a circuit to be made or broken. These switches ordinarily handle only auxiliary circuits.

A controller may comprise a number of units. These units must be so manipulated as to produce the desired connections and electrical conditions. There are many combinations possible which are not desirable and which will cause short-circuits or other improper conditions. In order to eliminate the possibilities for improper operation of the various units that make up a complete control, it is the practice to provide interlocks. These interlocks may take the form of mechanical connections, such as levers, preventing units from possible operation out of sequence. They may also take the form of auxiliary electrical contacts which serve to open certain circuits when other circuits are closed.

Interlocks may be provided between the units of a single controller, as in the case of a control for a wound-rotor induction motor where an interlock may prevent closing the primary circuit when the secondary resistance is cut out. Interlocks may also be installed to tie together separate controllers governing motors handling related functions. For instance, an arrangement may be made to stop the motor handling the feed to a given machine if the process motor itself is stopped

The Use of Pulverized Coal

By H. A. Schaffer
Conservation Engineer, Portland Cement Association

Experience gained through long use of powdered coal in the process of cement manufacture, especially with reference to handling, should prove useful in the application to power-plant work. It is the author's belief that the greatest development of power in the cement industry will lie along the lines of waste-heat utilization.

IN CONSIDERING the adoption of powdered coal as fuel for power plants, industrial executives often hesitate to shift from present practice because of the comparative newness of such installations in this field and on account of the reputed dangers linked with the use of explosive coal dust as fuel. Such men will find of interest the long experience of the portland cement industry with the use of powdered coal for burning cement clinker from the finely ground raw materials.

When rotary kilns for cement manufacturers were introduced in this country some 30 years ago, a change was necessarily made from the coke burned in the old-style stationary, vertical kilns. The new fuel had to be one that could be blown into the revolving cylinder to produce instantaneously the intense temperature needed for clinkering the raw materials.

At first crude oil was used with good results, but the increasing cost of this fuel in the East, where most of the early plants were located, led to trials of powdered coal. It was in 1897 that this finely ground material was first adopted as fuel for cement burning in several mills.

While crude oil and natural gas are today used in some localities where prices will permit, pulverized coal is the usual fuel. In fact, out of the 115 operating cement plants reporting to the United States Geological Survey in 1921, 92 burned powdered coal only and 6 more employed coal in conjunction with oil or gas. All but one of the remaining 17 burned oil; the exception utilized natural gas.

Kiln Fuel Requirements

Of course the fuel problems in the kilns of a cement mill are somewhat different from those met in power-plant practice. For instance, the great heat produced by a flame of pulverized coal is an advantage in cement manufacture. There the raw mix must be heated to a temperature of from 2,500 to 3,000 deg. F. in order to bring about the desired chemical reactions as the materials are burned into the clinker that is later ground into cement. But in a power plant this intense heat introduces difficulties, met to some extent by hollow furnace walls.

Ash, if reasonably constant and not too great in amount, offers no problem in the cement kiln, as allowance can be made for it in proportioning the mix. Apparatus for burning powdered coal in kilns is therefore simpler than that recommended for modern boiler installations, where arrangements for collecting ash must be made and attention directed to preventing injury to the furnace from heat.

In view of the long experience of the cement manufacturers with powdered coal in their kilns the question naturally arises as to why this fuel is not widely used in their power plants. The answer lies in the failures

that met their early attempts to secure the advantages of pulverized coal as a boiler fuel, and in the recent advances in the utilization of the hot kiln gases for heating the power-plant boilers.

Various experiments were made in power plants with the crude apparatus early available for burning powdered coal, but none of these trials was altogether successful. Probably the gravest error made lay in the failure to proportion the furnaces to suit the fuel, with a resultant inability to make them stand up in the hard, continuous service demanded in cement-mill operation. As many more pressing problems arose to occupy the manufacturers' attention and the possibilities of waste-heat utilization came to the front, the development of workable methods for burning powdered coal under boilers was left to others more vitally interested in that phase of power-plant design.

Renewed Interest Now Shown

Powdered-coal installations have recently been made in a few cement mills, yet it is hardly probable that a great deal more will be done in this field, simply because the engineers of the industry are working on an even more economical plan.

In most plants the proper conservation of the hot gases leaving the kilns will provide all the heat needed for the boilers in a power station big enough to supply the entire plant. As the power needed to operate the heavy machinery in a cement mill is large, this means a marked saving in fuel. The installment of waste-heat boilers requires a very heavy investment, however, and involves many problems that must be solved for each plant.

In any cases where additional fuel must be bought to generate power, the possibility of utilizing a low grade of coal when powdered is especially important, for a large quantity of fuel is required to make cement—about 200 lb. of coal to the barrel of 376 lb. when the waste heat is not utilized. Much more than half of this amount is now being pulverized for burning in the kilns, but a big total is still used without grinding to generate power.

In one case a cement plant has been able to burn anthracite silt, coke breeze and bituminous screenings in the same powdered-coal furnace, with satisfactory results. This furnace is equipped with both vertical and horizontal flare-type burners, for use with low- and high-volatile fuels, respectively. In the kilns it is difficult to use even a part anthracite, and the chemical composition of the coal must fall within certain limits.

Handling Powdered Coal

Especially in the field of safety the cement industry can offer valuable experience. In the early days coal dust was not generally recognized as a dangerous explosive, and inexperience resulted in a number of accidents. Since that time safe methods of handling this fuel have been worked out, and a campaign of education under the direction of the Accident Prevention and Insurance Bureau of the Portland Cement Association has greatly reduced the hazard.

Many plants have never had an accident from this source, and with the installation of modern equipment the industry as a whole is making the coal room safe. In 1922 the cement manufacturers of the United States burned over 5,800,000 tons of powdered coal in about 100 separate plants.

Above all, the grinding and conveying equipment should be tight enough to prevent the escape of dust. The coal buildings should be well lighted so that attendants can work with facility, and well ventilated so that there is good circulation of air. Under no conditions should open lights be allowed in the coal house. Magnetic separators should remove all iron from the incoming coal to avoid sparks in the grinders. The coal driers should be separated from the grinding equipment by a fire wall, at least. Experience has shown that danger has been greater to life than to property where accidents have occurred, as the trouble has been caused more by "flares" that severely burn workmen than by explosions that wreck buildings. Employees must be made to understand the necessity of continuous observations of established precautions, and none but trustworthy men should be employed in the coal department. It is not enough to post accident-prevention rules—they must be observed.

For cement burning, finely ground coal has been found essential as much heat is lost from incomplete combustion when coarser material is used. Eighty-five per cent passing a 200-mesh sieve is considered good. Fine grinding, however, tends to shorten the flame and to make it hotter, so that again the best boiler-room practice may differ from that cited.

In cement mills the procedure is first to dry the coal in rotary driers and then to pulverize it, usually in mills of the centrifugal type or in tube mills after any lumps have been broken up in small rolls or crushers. In a number of cases powdered coal is burned in the coal driers, but this is not such common practice as it is to use this fuel in drying the raw materials before they are ground to a fine powder. In at least one public-utility power plant the driers have been omitted entirely. The practice of pulverizing the material without drying is followed in one or more other cases when coal having a low moisture content is received.

Writing in the *Saturday Evening Post* not long ago, Floyd W. Parsons listed the cement industry as the fourth or fifth largest user of coal. Railroads come first, the iron and steel industry second, the public utilities third, and the domestic consumers fourth, if they are classed as a single group. Recent articles in *Power* have described the use of powdered-coal installations in public-utility stations and in a large iron working company's plant. In several countries the railroads have given attention to the use of pulverized fuel—notably in South America, where a number of locomotives operating on the Central Railroad of Brazil burn powdered fuel that is ground from a coal containing about 30 per cent ash and 9 per cent sulphur.

Advocates of powdered coal as fuel have therefore forced the entering wedge into the ranks of the first three industrial consumers of coal, and the fourth largest has long been the biggest user of powdered coal in the country.

Combined impulse and reaction turbines are smaller and require less material than solely reaction-type turbines. In the former steam is expanded from a nozzle and then passed through two rows of revolving blades, constituting an impulse stage where no expansion, practically speaking, takes place in the blades. The steam from this point is expanded in a long series of reaction blades, where some expansion takes place in each row. The impulse stage has fewer blades and takes up less room than a corresponding staging of the reaction type.

A New Weighing Meter for Liquids

The Cochrane weighing meter, an interesting device for automatically weighing condensate, feed water, etc., is adapted to the small plant where the use of a V-notch meter would be uneconomical. It may be built either for use with an open tank, as in Fig. 1, or incorporated with the open type of feed-water heater.

The principle of this meter is that of the ordinary beam scales. A bucket *B* is carried on trunnions on one end of the beam *A*, while at the other end are adjustable counterweights *WW*. The beam itself is fulcrumed upon knife-edges at *F*. A stop *S* determines the normal position of the beam, with the bucket empty. The operation is as follows: (1) Flow from the reservoir overhead is caught in the bucket until the latter has filled to a predetermined point.[1] (2) The instant that the weights are overbalanced, the bucket end of the beam sinks just enough to trip the latch *L*. (3) The bucket then tilts and dumps its contents, rolling on its trunnions, the flow being caught meanwhile in detaining chamber *C*. (4) As the bucket recovers, chamber *C* discharges its contents into the main bucket. A baffle prevents spilling, as shown. (5) In recovering, the bucket travels a short distance past the latch, and then, as more water flows in, slowly returns until it gently strikes the latch, without shock or jar. (6) The cycle is thus completed, with the bucket in the original position, ready to tilt when full, as before.

This meter, a product of the H.S.B.W.-Cochrane Corp., Philadelphia, Pa., is easily calibrated by weighing a single discharge. It is accurate within one per cent at all rates of flow. The amount discharged may be changed by moving the weights. If the flow is to be controlled after it has passed the meter rather than before, a float in the receiving tank may be connected to a valve *V* in the incoming line, as shown in Fig. 1. The bucket trunnions roll upon plane surfaces, with practically no wear. A record of the dumping of the bucket is obtained through the use of a counting train.

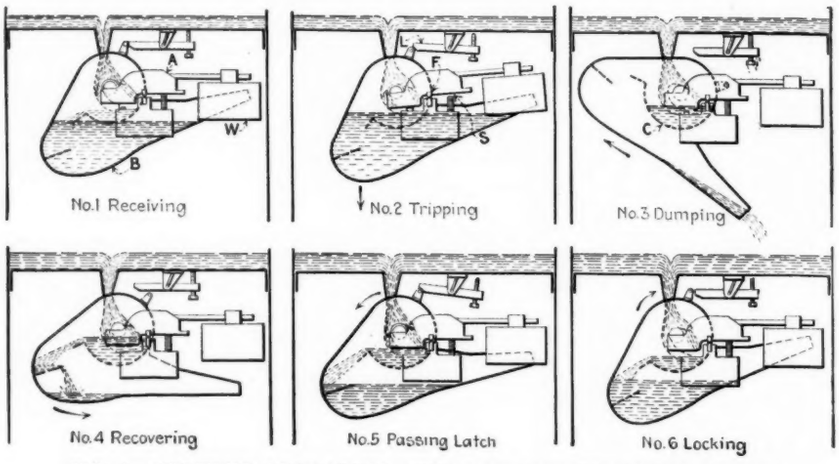


FIG. 2—SHOWING OPERATING CYCLE OF THE WEIGHING METER

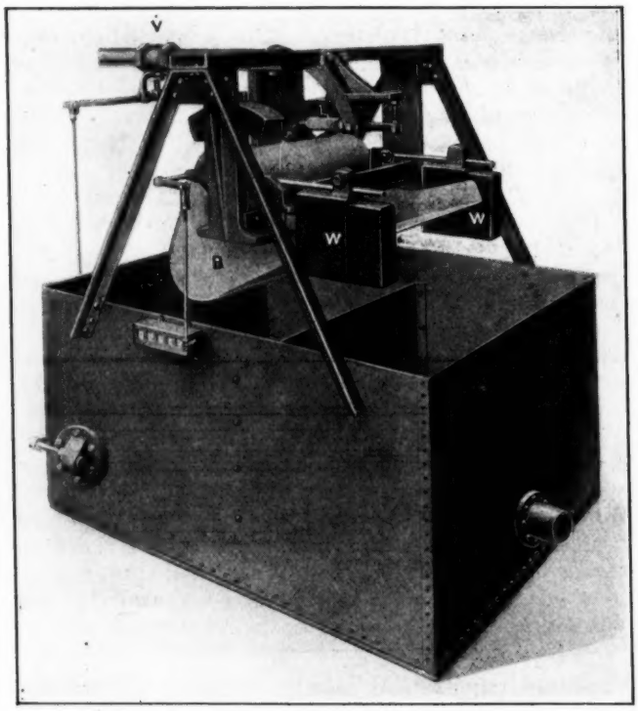


FIG. 1—COCHRANE WEIGHING METER, WITH OPEN TANK

[1] The figures in parentheses refer to the corresponding diagrams of Fig. 2.

The Unit of Heat

The standard unit of heat in this country is the B.t.u. (British thermal unit) which is the average amount of heat needed to raise the temperature of one pound of water one degree F. over a range of temperature from 32 deg. F. (melting point of ice) to 212 deg. F. (boiling point of water under standard atmospheric pressure).

The definition gives it as the average over this range, because the exact amount of heat required to raise a pound of water one degree F. varies a little at different temperatures. This variation, however, is so small that we need pay no attention to it in ordinary combustion problems, and can assume, with sufficient accuracy, that *one B.t.u. is the amount of heat required to raise a pound of water at any temperature one deg. F.* To raise one pound of water 5 deg. F. therefore takes 5 B.t.u., and to raise 6 lb. of water 50 deg. takes $6 \times 50 = 300$ B.t.u.

Through an unfortunate circumstance there have been in use in the petroleum industry two arbitrary scales for hydrometers; one based on the modulus 140 and the other on the modulus 141.5. This has led to much confusion and uncertainty. In order to clarify the situation and put the oil industry on a definite basis, the Bureau of Standards has decided to use 141.5 as the modulus, and in circular No. 19 there will be included a table giving Baumé values based on the formula,

$$\text{Degrees Baumé} = \frac{141.5}{\text{Sp.Gr. } 60°/60°\text{F.}} - 131.5$$

The United States Baumé scale based on the modulus 140 will continue to be used for other liquids lighter than water.

Reducing Refrigerating Power Costs

By M. S. Howe

Many factory managers devote their entire time to producing the article manufactured and ignore the power-plant end. Usually, the power plant is considered as a necessary evil to be kept in the background as much as possible, ignored save only when damning the high cost of the power used. It was my experience about two years ago to be confronted with a striking instance of this way of overlooking the economies possible to obtain in the production of power.

I was temporarily out of employment, and upon being offered the place of engineer in a Southern candy factory, accepted it with some degree of satisfaction. Upon my arrival I found the power plant of the factory to be mostly of a refrigerating nature. There were two 72-in. by 18-ft. horizontal tubular boilers furnishing steam at from 25 to 50 lb. pressure to the many candy-cooking kettles in the factory. As usual in all candy factories, a certain amount of refrigeration was required in the stock and chocolate rooms since the candy must be kept at a fairly low temperature after it is made; many candies, such as the chocolate-coated stock, must have the centers chilled enough to stay solid when run through the chocolate-dipping machine.

In addition to this customary load the temperature in New Orleans is so high that the several workrooms were cooled by a current of chilled air. The total

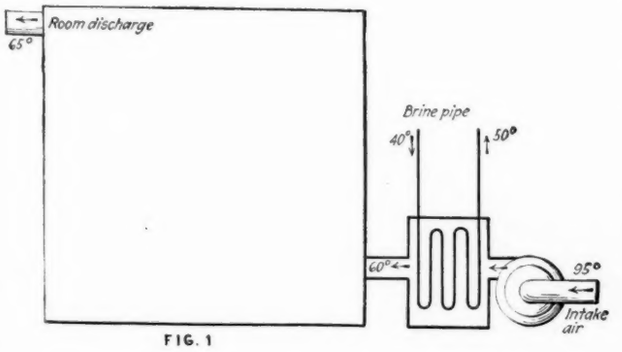


FIG. 1

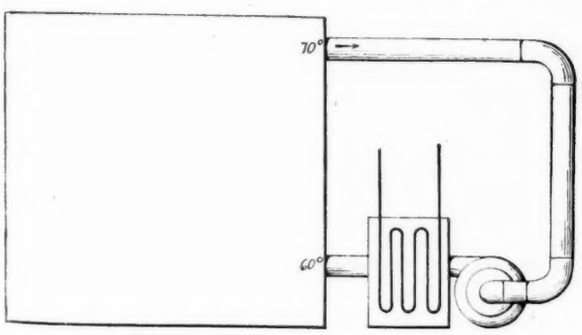
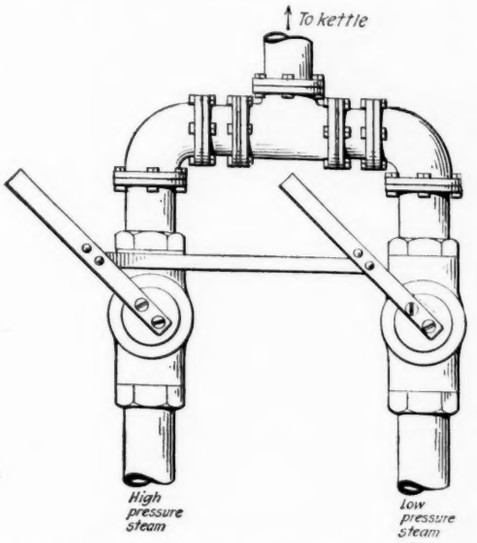


FIG. 2

FIGS. 1 AND 2—PAN OF COOLING SYSTEM

refrigerating load was carried by a 35-ton ice machine, driven by a 75-hp. motor, power for which was purchased from the electric-service company at five cents per kilowatt-hour. The total monthly electric-power bill, covering the compressor motor, a brine-pump motor, and a 3-hp. fan motor ran from $600 to $700 a month, the elevator and factory motors being on a separate meter.

The job was quite easy, as there was a negro fireman to care for the boilers and a night watchman who got steam up in the mornings. My duty was mainly with the refrigerating machine, for since it was in the



FIGS. 3—ARRANGEMENT OF HIGH- AND LOW-PRESSURE STEAM LINES

middle of the summer, with an outside temperature of 96 to 100 deg., the candy workers were continually kicking about the room temperature, which often ran above the desired 65 deg. On top of this the manager was seldom off the subject of the high cost of power. He considered the electric-power bill together with the two tons of coal for cooking steam entirely too much for the output of candy. Considering that these items, together with the wages of myself and the fireman, ran over fifty dollars a day, I could not help thinking he was right, and so I decided, even though the job was a temporary one for me, to dig in and see how this cost could be reduced. Needless to say, a cut in wages did not appeal to me and so any saving must come from a decrease in the coal or electric power costs.

Method of Air Cooling

The first thing that struck me as being extremely unreasonable was the system of taking in outside air at 90 to 100 deg., cooling it to 60 deg. before putting into the workrooms, and then discharging it at around 65 to 70 deg., as shown diagrammatically in Fig. 1. I could see no reason why this spent air should not be taken back through the cooling coils and used again. This meant a duty of lowering the temperature from 70 to 60 deg. instead of from 95 to 60 deg. Since there was close to 1,000 cu.ft. of air handled per minute, the cooling of the air required the removal of $1{,}000 \times 1/13 \times 0.25 \times 35 = 700$ B.t.u. (the specific heat of air is 0.25, the temperature drop is 35 deg. and a pound of air equals 13 cu.ft.). This was for cooling the air. Now each cubic foot of air drawn in by the blower takes with it a certain amount of water vapor, depending on the humidity. With a humidity of 70 the 90-deg. air on entering the cooler contains 1.6 lb. of water vapor per 1,000 cu.ft. This water vapor is chilled to 60 deg., as is the air, and each degree reduction in temperature requires the removal of 1 B.t.u. per

pound, or the water-cooling amounted to 56 B.t.u. Including the air cooling the total heat removal was 756 B.t.u. per minute, or equal to a duty of approximately 4 tons of refrigeration, requiring about 8 horsepower.

A sheet-steel pipe with branches from each of the workrooms was run to the fan, as indicated in Fig. 2. The air to be cooled was taken from the rooms at from 68 to 70 deg., and in this way the reduction in air temperature was from 70 to 60 deg. instead of from 95 deg. The refrigeration duty was then reduced over two-thirds. By this alteration, which cost less than $200, the saving in refrigeration was about 3 tons; this meant over $100 a month.

Saving the Condenser Water

This change opened the manager's eyes, and he began to want to know how some more money could be saved. It was not difficult to suggest to him that, instead of allowing the cooling water from the condenser to go to waste to the sewer, he put in a cooling tower on the roof. Since the water was costing over $150 a month, the cost of the tower would be made up in a few months; the cost of the makeup water and of pumping after the condenser was put on the roof was but a small part of the former water cost.

The manager began to neglect the candy-selling end and hunt for ways and means of reducing the costs all over the plant. One could almost see his mouth water when a scheme of reducing expense was broached. In fact, although I still looked upon the job as being merely a way to live until a more promising one opened up, the game soon got me, and I often found myself lying awake at nights working up schemes. One of the things that was always on my mind was the large electric-power bill considering how little the plant power was.

After studying the matter from every angle, I decided that it would be a good thing if a steam engine was installed to drive the compressor, doing away with the motor. It seemed feasible to me to use the exhaust steam from the engine to supply the candy kettles. The manager shook his head dubiously at this; candy making, so he said, required high-pressure steam and he was sure the low-pressure exhaust would fail to do the cooking.

Installation of Engine

Now the steam pressure was seldom higher than 50 lb. gage, which meant a temperature of 298 deg. F. If a back pressure of 10 lb. was carried on the engine, the temperature would be 250 deg., and I felt that the 48 deg. difference would not seriously hurt the cooking process. With these arguments the manager, since he had the saving bee, was convinced, and as a result an 85-hp. four-valve engine was installed. The manager wanted a cheaper single-valve outfit, but I was afraid that the engine steam consumption, if it was high, would exceed the cooking-kettle load. The engine was belted to the compressor, and to provide electric power for the fan, brine motor and a few small motors, a second-hand generator was installed and belted to a split pulley on the engine shaft. Since it was necessary to have some kind of breakdown service, it looked as if we would have to keep the connection with the electric-service company. The rate for breakdown service was unreasonably high, so a 25-hp. kerosene engine costing $350 was put in, together with a 20-kw. generator we picked up for $200.

Altogether we had spent close to $3,000, and the manager began to have some doubts as to the wisdom of the change. He was certain the cookers would not be able to use the 10-lb. steam. This prompted me to suggest that when we made the change the kettle gages be set up to read 35 lb. with a 10-lb. pressure. I thought that this would fool the men and they would not be able to object to the change.

After everything was ready, the engine was started on Sunday morning for a test. It handled the compressor and the other loads without trouble. When Monday morning came the candymakers went to work without knowing that they were using the engine exhaust steam. The 35 lb. on the gage caused some kicking, but things ran along pretty well. We found that the low-pressure steam did slow up the cooking, so a live-steam line was run to the cookers and the men were ordered to use the low-pressure steam until it became necessary to raise the kettle temperature, when the live-steam pressure valve was opened and the low-pressure one closed. It was found that sometimes the men forgot to close this valve. This trouble was eliminated by placing these valves, which were of the plug kind, close together and fastening the handles together with a cross-bar, as shown in Fig. 3, so that one valve could not be opened unless the second one was closed.

Reducing Use of High Pressure Steam

There was a tendency for the men to use the live steam too much of the time, but this was stopped by placing in the manager's room an electric bell that rang every time the back pressure went high enough to raise the atmospheric back-pressure valve on the engine exhaust line. In this way the men were induced to see that the low pressure was used as much as possible.

After little adjustments were taken care of, the system worked out as well as was hoped for. The coal consumption increased about 1½ tons, but the entire electric-power bill was eliminated. The cost of operating the power plant was $26 a day, including the wages of myself and the fireman. The lubricating oil, waste and such stuff did not run over $10 per month. The manager figured that the saving was between $30 and $35 a day, which meant $750 a month. Since the candy output was not so large as that of some factories, no doubt this direct power saving more than equaled the one cent a pound which is advertised as being the total profit on candy of a famous maker.

Shortly after the new equipment was put into first-class shape, I left to get back into my regular line. I rather regretted this, for there were so many ways in which power and heat could be saved that I really wanted to see it through. For example, the length of the cooking is mostly a matter of guesswork by the men. If a measuring device were placed on the trap from each cooker, a record could be kept of the quantity of steam used per kettle and the men paid a bonus for any saving.

Since the resistance of the fuel bed is ordinarily the only changeable resistance in a boiler-draft problem, it may be said that the weight of gases varies inversely as the resistance of the fuel bed, the total pressure drop remaining constant. Less heat is delivered to the boiler when the resistance through the fuel bed increases, the initial temperature of the gas and the total pressure remaining constant.

EDITORIALS

F. R. LOW, EDITOR

How Much Heat?

THE height of a building is one thing, the elevation of its roof above the sea level is another. The only possible way to figure the height of a building from elevations is to take the difference between the elevation at the top and that at the bottom.

This illustrates a mistake that engineers frequently make when talking about heat. All substances contain heat—and a lot of it—even when the temperature is forty degrees below zero. The steam tables give the heat content of water at thirty-two degrees Fahrenheit as zero. This, of course, is entirely arbitrary but it finds its practical justification in the following facts: First, there is no way of ascertaining the total amount of heat the water contains measured above absolute zero; second, engineers are not interested in actual heat contents, but only in heat differences; and third, the freezing point of water is a convenient reference level for most problems in steam engineering.

Any "heat content" given in the steam tables is merely the heat required to raise one pound of water from thirty-two degrees to the given condition. All heats measured above this point are arbitrary, just as elevations measured above sea level are arbitrary. Only the difference between two elevations has meaning in terms of height; in the same way only the difference between two "heats" has any practical meaning in engineering. The "heat" in one pound of water at 190 degrees Fahrenheit is 158 B.t.u. according to the steam tables. Yet this figure, as it stands, tells us no more about the heat lost when one pound of 190-degree water is wasted than we could learn about the fall from the roof of a building from the fact that the roof was 158 feet above sea level. We must know the elevation of the ground as well as the roof to see how far a man could fall. In the same way we must know the heat content per pound of the available water supply from which the 190-degree water would be produced. Subtracting this from 158 B.t.u. gives the true fall, or loss.

Where the Difficulty Lies

ALTHOUGH this country leads the world in applying machinery to man's service, we are one of the most wasteful nations of the world. Much has been done toward remedying this condition, but there still remains an appalling waste, not only in materials but also in human endeavor. Yet some of our industrialists would have the bars against immigration lowered to flood the country with cheap labor so that the old order of waste may continue. This may be the easiest way out of the present shortage, but it certainly does not afford a solution of our industrial problem.

In some industries our wasteful methods make it necessary to use twice as many workers as would be required if properly employed. Although the coal industry is one of the worst offenders in this respect, it is by no means the only guilty one. Furthermore, in some cases conditions are getting worse instead of better. In 1921 the average number of days worked by the coal miners of this country was one hundred and forty-nine, or less than fifty per cent of the working year. In that year less coal was mined than in 1910, although over one-hundred thousand more miners were employed. It is estimated that there are over two-hundred thousand more miners in the coal industry than are required.

This waste of labor is not all due to conditions for which the coal-mining industry is responsible. The transportation and distribution of coal is done most inefficiently. In the summer months, when coal can be mined, transported and distributed easiest, production is at a minimum, while in winter mining and transportation have to be maintained at a maximum to supply the demand. With cars enough to handle more than twice the coal required by the country, there is a car shortage each year. If mining and transportation problems were properly solved, not only would there be sufficient cars and adequate distribution facilities, but thousands of workmen could be released for other industries and those required in the transportation of coal could be given steady employment.

The inefficient use of fuel places a heavy unnecessary tax on the labor supply. Electrification of the trunk-line railroads would cut the fuel consumption in this industry to about one-third what it is with steam operation. Many power plants are also serious offenders in this respect. What is true of the fuel and power industries is equally true of many others. Then again, there is the waste of labor due to accidents, which is estimated at about three hundred million working days per year; in other words one million people are continually prevented from following their vocations. It is claimed that eighty per cent of these accidents are preventable, which would be equivalent to providing eight hundred thousand men and women for industry. Owing to illness and non-industrial accidents another million are continually kept from their employment. Estimates of this condition indicate that over fifty per cent is preventable.

Then there are our periods of business inflation and deflation, causing labor shortages followed by periods of unemployment. During 1920 and 1921 millions of wage earners were unemployed and factories representing tremendous investments were closed, resulting in heavy losses both to labor and to investors. Now a shortage in production and labor is being experienced. Although it may be impossible to eliminate these conditions entirely, very little or nothing has been done to effect even a partial remedy. Recently, Secretary Hoover has made proposals that have for their purpose the flattening out of the alternate cycles of abnormal prosperity and resultant deflation, but, it is too early to predict what the outcome of this effort will be. What is needed in this country is not more labor, but a better direction of what we have, and this is one of the big problems that the engineer and those responsible for future development must undertake.

Steam Turbines Well Suited to Higher Pressures and Temperatures

THE reliability of large steam-turbine units of a few years ago was not all that could be desired. Far-reaching improvements, however, now represent a happier state of affairs. The advent of higher pressures and temperatures, on the other hand, may be viewed with alarm, when the effects of increased expansion, decreased tensile strength, requirements of greater flexibility, etc., are considered, in connection with large units. Increased strength for high-pressure elements must be provided.

Summing up the factors, however, indicates that the turbine is readily adaptable to more severe steam condition. For extremes of pressure, a reliability equal to or greater than present large single-cylinder types may be attained.

Higher-temperature steam would certainly affect turbine blades adversely, if the temperature were raised to the point where tensile strength is sufficiently impaired. But when pressure and temperature are both increased the proper relative amount, the first expansion can be arranged efficiently in a stationary nozzle, so as to supply steam of normal temperature to impulse blades. The impulse stage therefore may be utilized to control the most vital factor, which is temperature, by confining high temperature to the piping and high-pressure nozzle chamber. In some of the latest turbine designs such a chamber is mounted separate from the cylinder except for points of attachment or guides.

Blading, comparatively speaking, constitutes the delicate part of a turbine. The capacity of efficient large turbines is limited by the exhaust-blade area. This in turn depends on the strength of materials, so that such blading already operates at the safe limit of strength of the metals or alloys available.

In high-pressure blading, however, there is a great reserve of strength. Steam volumes are so much smaller at these points that the short blades have to resist relatively very little centrifugal force. Even high temperatures in direct contact with such blades would not necessarily be disastrous at present blade speeds. It would be necessary for the temperature to reduce tensile by an amount equal to the reserve that is now available, in order to reach the danger point.

As pressures advance, the reheating cycle becomes more advantageous in many ways. At one thousand to fifteen hundred pounds the cross-compound type tends to meet with favor where a high-pressure impulse element is utilized for the first expansion, and the exhaust steam is reheated before being supplied to the lower-pressure turbine. Thus two or more turbines replace the large single-cylinder unit of the present type, and there is greater opportunity of localizing trouble. It is customary to arrange for the operation of any element independently of the others; the entire cross-compound unit is therefore less likely to be incapacitated and represents a greater degree of potential reliability.

Higher temperatures and pressures require the development of special valves. Difficulties here multiply rapidly as steam conditions become more severe. Since a one-thousand-kilowatt unit at fifteen hundred pound steam would require a high-pressure steam line about an inch and a quarter inch in diameter, the valve sizes at least would be greatly reduced, and suggest that other principles than now employed for steam control might be worth investigation.

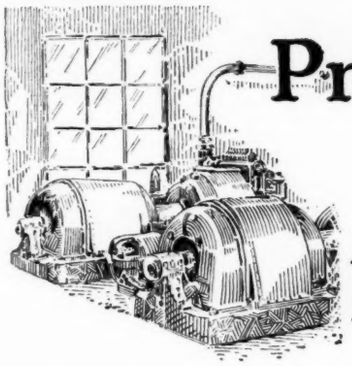
Feed Water for High-Pressure Boilers

MANY signs indicate a trend toward the general adoption of higher steam pressures, particularly in the large plants, but when it comes to predicting the average pressure carried ten years hence, one guess is as good as another. It may be 300, 500, 800, 1,000 or even 1,500 lb. All these pressures are mechanically possible, and the point of best commercial economy may be shifted up or down at any time by new developments that change the relative cost and thermal efficiency of plants designed to operate at the various pressures. Whatever this optimum pressure may prove to be, it will probably be well above the average now prevailing. More than this it is unsafe to predict.

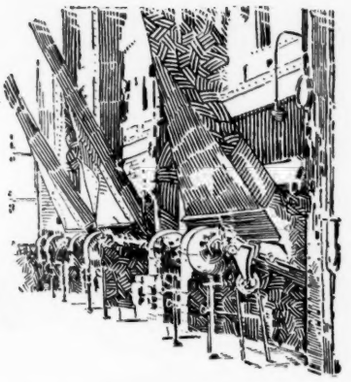
The general adoption of higher pressures will certainly place additional emphasis on the matter, already highly important, of feed-water purification. If a boiler is to be operated at one thousand pounds pressure, scale must be taboo. The temperature of a clean tube at this pressure would be around 550 deg., as against about 390 deg. for a pressure of two hundred pounds. With a given thickness of scale and a given rate of heat transmission the high-pressure tube would still remain about 160 degrees hotter than that of the "low-pressure" boiler. Since the lower banks of tubes in present-day boilers are subject to bagging in the presence of fairly thin layers of scale, it would be expected that a mere film of scale would produce the same effect at the higher pressure. Moreover, the smaller tube bore naturally associated with extremely high pressures would increase the difficulties of scale removal. In view of these considerations it would seem that the safe operation of a one thousand-pound boiler would require the complete elimination of all scale-forming impurities from the feed water.

Another characteristic of high-pressure steam generation is the increased tendency to foam and prime. This is due in large measure to the fact that the density of steam increases rapidly with rising pressure, while the density of water decreases. At a pressure of two-hundred pounds water is more than one hundred times as dense as saturated steam, while at one thousand pounds it is only twenty times as dense. As a natural result of this smaller difference in density "gobs" of water are more readily carried over into the steam main. This tendency to prime is made more serious by the smaller tubes and disengaging drums characteristic of high-pressure boilers.

In view of these conditions it appears that the feed water for a one one thousand-pound boiler should be free not only from all scale-forming impurities, but also from any substances that might assist foaming or priming. To meet these requirements, it would be necessary to feed such a boiler with pure distilled water. This would present no difficulty in the case of a condensing plant with a small percentage of makeup, since evaporators are now operating with commercial success in many condensing plants with ordinary boiler pressures. The requirement of distilled feed water might, however, be a factor limiting the use of extremely high boiler pressures in isolated plants where a large quantity of exhaust or bled steam (other than coil heating in which the condensate is recoverd) is used for process work. With a make-up running up to one hundred per cent the cost of distillation would become prohibitive.



Practical Ideas *from* Practical Men



What Caused the Freak CO_2 Readings?

For several years past I made numerous observations and tests of refuse-wood boiler furnaces in Washington. Oregon and British Columbia. At times when taking Orsat readings freak results that could not be accounted for were obtained. These discrepancies usually took the form of high CO_2 percentages with CO_2 plus O_2 plus CO percentages well over the calculated values.

While making tests of a new type of furnace with novel grate and refractory positions, some freak Orsat results were obtained and, conditions being favorable, an effort was made to determine their cause.

The fuel was made up of hemlock, spruce and larch sawdust, hand barker shavings and hogged waste wood, all byproducts of a paper-mill woodroom; average moisture 52 per cent, with an ultimate analysis approximating carbon 49 per cent, hydrogen 6 per cent, oxygen 45 per cent. With no excess air and with perfect combustion the fuel permits a theoretical maximum CO_2 of 20.5 per cent, the inherent hydrogen being almost completely oxidized by the inherent oxygen.

Samples were taken from the first pass of the boiler in order to eliminate the setting leakage variable. All samples were cooled with saturated water, uncooled samples usually reading from 1½ to 2 per cent higher than those cooled. Temperatures of solutions and samples were kept at 70 deg. Solutions were frequently renewed and checked against air blanks and standard CO_2. All sources of leakage were eliminated and checks for tightness of cocks and other fittings were made.

The furnaces were watched for a number of days, and whenever conditions pointed to possible abnormal readings, Orsat determinations were made. On March 20 some interesting results were obtained, the following compilation including the more important determinations made:

No.	Per Cent CO_2	Per Cent CO_2 Plus O_2	Per Cent CO_2 Plus O_2 Plus CO	Time
1	17.0	20.4	20.6	10.30 a.m.
2	18.0	20.8	20.8	10.45 a.m.
3	20.0	21.2	21.2	11.30 a.m.
4	20.4	21.5	...	11.37 a.m.
5	20.0	22.5	...	2.50 p.m.
6	17.4	...	...	2.55 p.m.
7	19.0	20.8	21.4	3.00 p.m.
8	23.0	25.6	25.6	3.50 p.m.
9	23.0	23.6	23.8	4.00 p.m.

A number of informal samples taken between eleven and twelve o'clock tested 20 to 21 per cent CO_2 with a maximum of 22.5 per cent CO_2 plus O_2. Simultaneously with the No. 9 sample two samples were drawn for a titration test. One-tenth and one-twentieth normal KOH solutions were used for absorption. Methyl orange was used as the indicator. The titration results gave an average of 22.6 per cent CO_2, which is a rather good check against the 23 per cent Orsat determinations.

On March 23 further results were obtained as follows:

No.	Per Cent CO_2	Per Cent CO_2 Plus O_2	Per Cent CO_2 Plus O_2 Plus CO	Time
1	11.0	19.0	19.6	2.55 p.m.
2	7.6	20.0	20.0	3.05 p.m.
3	12.8	19.2	19.6	3.15 p.m.
4	11.2	19.8	20.0	3.30 p.m.
5	8.0	19.2	19.2	3.40 p.m.
6	14.2	17.6	20.4	3.50 p.m.
7	15.8	16.6	19.4	4.00 p.m.
8	19.6	20.4	20.8	4.15 p.m.
9	18.6	19.2	19.2	4.35 p.m.

These tests were made with variations of draft, auxiliary air and fuel feed and are offered as an example of every-day results. As far as could be determined, the fuel was identical with that giving the high CO_2.

A study of the results seems to indicate the presence in the fuel of either inherent CO_2 or some other compound that has similar properties. Several authorities on the chemistry and structure of wood were consulted, but the various complex formulas for wood fiber and lignum presented failed to show the presence of any such compound. I should be pleased to have the matter discussed and should like to hear of others who have made similar investigations. M. S. GEREND.

San Francisco, Calif.

Packing Causes Engine to Pound

A symposium by some of the old operating men on what can make an engine pound would make interesting reading. The most peculiar and difficult pound to locate I ever had to do with in several years' erecting work was caused by piston-rod packing.

The engine was a simple Corliss, 14x32-in. 110-r.p.m., operating non-condensing. It had run about a year when the purchasers reported that a severe pound had developed and they wanted a man right away. I happened to be the nearest man. Sure enough, the engine would run quietly awhile and on a slight increase in load would pound sharply when turning the crank center. It sounded much like someone striking the cylinder with a sledge.

The usual inspection of piston, crosshead pin, crankpin, etc., was made and everything found to be in good shape. Indicator cards were taken and compression experimented with, but the pound was still there. I noticed that the rod did not run quite level, the crosshead end being too high. I readjusted this with the result that the piston rod heated up and the pound became more violent. I was informed that metallic braided packing had been installed some weeks previously. This was in rings that were used the same as soft packing. The engineer had kept tightening it until it was like solid rings of babbitt and was rigid enough and had bearing surface enough to lift the piston as it

turned the crank center. The packing was taken out, soft packing installed, and the pound disappeared.

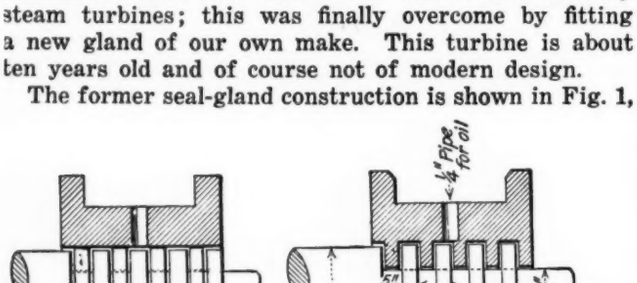
It has been my experience that some engines require lots of compression, while others will run smoothly with little or none. Pumps and compressors require less because the water or air takes up the momentum of the parts. I had a pump that ran smoothly with wedge bolts two turns slack (about 0.025 in.) on the crank brasses. Some will pound if they get more than oil clearance between brasses and pins. J. R. BELKNAP.

Lansing, Mich.

Improvement in Small Turbine Gland

The company with which I am connected had considerable trouble with the seals on one of the auxiliary steam turbines; this was finally overcome by fitting a new gland of our own make. This turbine is about ten years old and of course not of modern design.

The former seal-gland construction is shown in Fig. 1,



FIGS. 1 AND 2—ORIGINAL AND IMPROVED TYPES OF GLAND

in which rings were fitted between grooves in the shaft. This arrangement worked with reasonable satisfaction on low back pressure (1 to 3 lb.), but under pressures of 5 to 8 lb. the steam blew out along the shaft, condensed, and floated the oil out of the bearings. It became necessary to fill the bearings with oil as frequently as every half-hour.

The gland was rebuilt as shown in Fig. 2, using a smooth shaft with rings revolving in grooves in the housing. This seal operates with entire satisfaction at 10 lb. back pressure. T. N. LYTTLE.

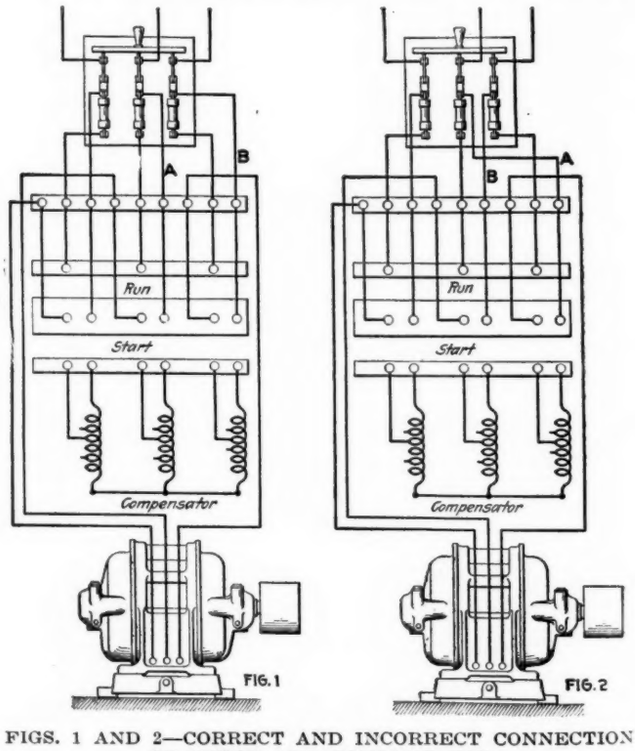
Springfield, Ohio.

Motor Started Three-Phase and Ran Single-Phase

A hurry-up call from a grain elevator brought me to the place, and on inquiring as to the behavior of the machine, I was told that a 40-hp. 440-volt 60-cycle three-phase 720-r.p.m. squirrel-cage motor, which had run satisfactorily for a period of six months, had suddenly developed a bad case of blowing fuses. The starter for the motor was on the ground floor, while the motor was on the third floor. As the motor had run satisfactorily for a period of six months, it was taken for granted that the trouble was in the motor itself, but upon inspection of the winding and bearings and testing for grounds, it was apparent that the machine was all right, as there was no visible cause of trouble.

It was decided to try starting the machine, and when the switch was thrown to the starting position I could hear the motor come up to speed. As the speed became constant, I threw the compensator switch over to running position, but just then the fuses blew. On reloading them it was noticed that they were loaded with 60-ampere fillers. As the fuses blew almost instantaneously, it was decided to double them and put two 60-ampere fillers in each. After taking the belt off, the motor was ready to try again. This time it came up to speed as before, and when throwing to running position a loud growl was heard and then smooth running. Leaving the starter on the running position and going up to the motor, I found that it was running as smoothly as any motor could be expected. However, what caused the growl when throwing from the starting to the running position was a mystery. After giving a man instructions on how to proceed in starting, I went up to the motor and waited for it to be started. After starting and coming up to speed, it slowed down quickly, stopped and started in the opposite direction. The cause of the trouble was plain—the motor received different phase rotation for starting from that received when running. Upon tracing the wiring, which was in conduit, and reversing the proper connection, reloading the fuses with 60-ampere fillers, putting the belt and load on the motor, the machine was started, and this time it behaved normally.

The explanation of the trouble was that the wireman had crossed the wires between the switch and compensator, and upon starting the first time only one fuse blew and the machine continued to run as a single-phase motor. After this occurrence it started three-phase with the fuses out of circuit and ran on one phase with the fuses in, as shown in Figs. 1 and 2. In Fig. 1 the



FIGS. 1 AND 2—CORRECT AND INCORRECT CONNECTION OF MOTOR AND COMPENSATOR

In Fig. 2 wires *A* and *B* are crossed from the correct connection in Fig. 1, which will cause the motor to start in one direction and when the switch is closed to the running position the motor will reverse or blow the fuses.

connections are correct, while in Fig. 2 wires *A* and *B* are crossed, which would cause the motor to start in one direction and when the switch was closed to the running position it would run in the opposite direction.

Denver, Colo. ANTHONY J. MORRONI.

Comments from Readers

Idle Boiler Should Be Cut Off from Circulating System

In the March 13 issue, there is an article entitled "Destructive Heating-Boiler Explosion," giving an illustration of two hot-water heating boilers connected together with shutoff valves in the discharge and return lines, but with a common header on the outlet between the boilers and a common return between the return valves.

In the fifth paragraph of the article, it is stated: "There is no advantage in placing stop valves in any of the pipes connected to a hot-water heating boiler, for if there is more than one boiler connected to the system and the service of only one of them is desired, there is no reason why the idle boiler should not remain filled with water and in direct connection with the system."

As a matter of operation I think any engineer would disagree with this statement, as there certainly would be a loss to have an idle boiler included in the circulating system. Besides this, without the valves referred to, it would be impossible to make repairs to either boiler with the other in service. The only element of danger which may exist and, apparently, is not specifically provided for, would be taken care of either by a safety valve on each boiler or by an independent connection below the boiler stop valve to the expansion tank.

Newark, N. J. G. E. LONG.

Three Years' Progress

In the March 6 issue of *Power* there appeared an editorial entitled "Three Years' Progress," figures being given showing the coal consumption and kilowatt output of the public-utility power plants in the United States as indicated by data from the United States Geological Survey and expressing regret that corresponding data are not available for the industrial plants of the country. The following description of an industrial plant and its data covering fuel consumption per unit of output is given with the hope that some other plants may be induced to present comparative data.

The plant in question consists of two separate units, each with its battery of boilers and operating at different pressures. The older of the units consists of 1,500-sq.ft. horizontal return tubular boilers operating at 150 lb. gage, saturated, furnishing steam for a vertical cross-compound engine of about 1,600 horsepower.

The other unit consists of 5,000-sq.ft. water-tube boilers fitted with superheaters and operated at 170 lb. pressure. This battery of boilers furnishes steam for a 2,000-kw. turbine and steam-driven auxiliaries.

In order to get a more flexible and economical arrangement, two of the old horizontal return-tubular boilers were equipped with superheaters and connected into the same steam line as the water-tube boilers. These two boilers and one water-tube unit furnish sufficient steam for the turbine which is running at a slight overload.

Eight of the horizontal return-tube boilers are sufficient to furnish steam for the engine unit and auxiliaries, and in addition all the steam used for heating and a small amount for process work. The fuel per horsepower-hour as charged to the engine unit includes the fuel used for these other services.

All the boilers are oil-fired except two, one of which is equipped to burn factory waste only and the other to burn either factory waste or oil. The boilers are equipped with the necessary draft gages, recording thermometers and CO_2 instruments.

The oil fuel has a heat value of 18,300 B.t.u. per lb., the waste fuel 8,700 B.t.u. per lb. All records are kept in terms of coal equivalent at 14,500 B.t.u.; that is, the total weight of oil consumed in a given period is multiplied by 1.26 to get the equivalent coal consumption, and the weight of the waste consumed in the same period is multiplied by 0.6 to get the equivalent coal consumption.

All factory waste is carefully weighed before being blown to the boiler room, and the total daily oil consumption is measured from the "service tank" from

OUTPUT AND EFFICIENCIES OF PLANT

Turbine Plant

Week Ending	Coal per kw.-hr. Kw.-Hr. Output	Coal per kw.-hr. 48-Hr. Run	Coal per kw.-hr. Including Standby	Per Cent Efficiency Operating	Per Cent Efficiency Including Standby
July 15, 1922	87,000	2.07	2.46	11.35 and	9.56
Nov. 18, 1922	93,320	2.05	2.41	11.45 and	9.76
Jan. 13, 1923	93,300	2.07	2.43	11.35 and	9.68
March 3, 1923	96,900	2.00	2.35	11.77 and	10.00

Engine Plant

Week Ending	Coal per hp.-hr. Hp.-hr. Output	Coal per hp.-hr. 48 Hr. Run	Coal per hp.-hr. Including Standby	Per cent Efficiency Operating	Per cent Efficiency Including Standby
July 15, 1922	76,092	1.74	2.03	10.08	8.64
Nov. 18, 1922	77,036	2.19	2.89	8.02	6.17
Jan. 13, 1923	78,347	2.36	4.21	7.44	4.17
March 3, 1923	78,417	2.48	3.55	7.08	4.94

Combined Plants

Week Ending	Coal per hp.-hr Total hp.-hr. Output	Coal per hp.-hr 48 Hr. Run	Coal per hp.-hr Including Standby	Per Cent Efficiency Operating	Per Cent Efficiency Including Standby
July 15, 1922	193,482	1.60	1.92	10.98	9.15
Nov. 18, 1922	202,109	1.76	2.21	9.96	7.95
Jan. 13, 1923	207,389	1.77	2.72	9.92	6.46
March 3, 1923	208,257	1.86	2.43	9.42	7.22

which the oil is pumped to the boilers, by means of a graduated depth gage. The service tank has two compartments, one being filled while the other is being emptied.

As a further check on the amount of fuel consumed, the main storage tanks are measured at weekly intervals and the amount used from storage is checked against the service-tank figure; the two figures seldom vary more than a fraction of 1 per cent.

The running time of the plant is 48 hours a week, which gives a "stand-by" of 120 hours a week, during which time a pressure of 100 lb. is carried on one or more boilers for fire protection and the necessary heating.

The output of the turbine unit is measured by a watt-hour meter, no account being made of power generated by the turbine auxiliaries. The power output of the

engine unit is determined by indicator cards taken each day, alternating a.m. and p.m.

The figures given in the table were taken at random, no attempt being made to get the results from the highest weeks, the desire being to show actual average conditions, and no correction has been made for unequal heating load.

It will be noticed that the turbine-plant efficiencies run quite uniform, but the engine-plant efficiencies vary with the heating load, since all heating is charged to the engine plant.

C. B. HUDSON.

North Plymouth, Mass.

Determining the Consumption of Steam Jets

I have noticed recently several articles on the use of steam jets under boilers, etc., and I wonder if the users of such devices have always realized the amount of steam required to operate steam jets.

According to the formula generally used for computing the flow of steam through orifices in thin plate, the flow of steam per hour, with 100 lb. gage pressure, is as follows:

Orifice Diameter, Inches	Consumption Lb. per Hour
1/32	4.5
1/16	18.
3/32	42.2
1/8	73.8

I have made a number of tests in the boiler room to determine the amount of steam used by steam jets. These results do not agree exactly with the theoretical—probably because the tests were made on holes drilled in pipe, and the holes were more or less rough, and with burred edges on the inside of the pipe. Anyhow, they show that the formula is not far from correct.

Hole Diameter, Inches	Steam Pressure Lb. Gage	Consumption, Lb. per Hour
3/32	100	52
3/32	70	35
3/32	60	36
3/32	40	22
3/32	40	19

From the data given, it will be seen that with a pressure of about 60 lb. gage the steam that escapes through a 3/32-in. orifice is equivalent to approximately one per cent of the normal output of a 1,000-sq.ft. boiler. I mention this size in particular, for it seems to be the size most commonly used in undergrate blowers. I have tested undergrate blowers that had twenty or more 3/32-in. holes and have found such devices to be using as high as 16 per cent of the steam generated.

The following method of testing for the amount of steam used by steam jets, small boiler-feed pumps, leaks through traps, etc., may prove of value, as it is simple and reliable.

In the case of a steam jet or jets, run a small pipe line to the line supplying the jets, and so arrange that the end of this line will be submerged three or four inches in a barrel of water placed on a platform scale. Put a valve on the line, also a gage to record pressures, and cap the end of the pipe where it is submerged under the water, and drill a hole in the side of the pipe about one inch from the capped end, so that steam may escape under the same pressure and through the same size of orifice as that used in the steam-jet device.

Now fill the barrel with water till the orifice is submerged about two inches, and balance the scales. Then turn on the steam, which will of course be condensed by the water as it issues from the jet. Note the increase in weight of water in the barrel for a given period, say fifteen minutes, and from this compute the weight of steam issuing from the orifice for one hour. Multiply this result by the number of orifices in use, and you have the total hourly steam consumption.

Be sure that all holes in the jet blower are clean and free from scale and that the test orifice is exactly the same size as the orifices in the blower that you are testing. Make your test orifice in the side of the pipe or in a cap placed on an ell at the end of the pipe, but not in a cap on the bottom of the pipe, otherwise it may fill with scale and give inaccurate results.

The same general scheme may be used in testing small steam pumps, leaky traps, etc., except that no orifice is necessary. Simply disconnect the exhaust and pipe it to a barrel of water on a platform scale, and submerge the end of the exhaust so that the discharged steam will be condensed, and note the increase in weight for a given period of time. I have used this method many times for field tests and find it simple and accurate.

There are on the market many devices using steam jets, and the usual answer to the query, "How much steam does the device use?" is "Practically nothing," or 1 per cent to as high as 5 per cent. If you are using steam jets, I would suggest that you make such a test as I have outlined.

E. E. HARPER.

Kansas City, Mo.

What Design of Furnace Is Best for Burning Mill Refuse?

The company with which I am connected as master mechanic operates a large sash and door factory and is contemplating extending the plant. We are at present buying electric power, and the question has come up as to the advisability of installing our own steam plant, as we are burning the sawdust and savings and other refuse from the mill in a burner with a great loss of heat that could be utilized in generating steam.

I would like to have some opinions from readers of *Power* who have had experience in burning waste fuel under boilers, as to the best type of furnace, arrangement of feeding, etc., also some comments on turbine-driven units in comparison with reciprocating-engine-driven units.

R. B. MCQUIDDY.

Independence, Kan.

Condenser Operation

In discussing the operating characteristics of ammonia condensers in *Power*, March 20, O. A. Anderson states that in the bottom-supply atmosphere condenser the hot ammonia vapor entering at the bottom coil becomes submerged and condenses in a body of liquid ammonia. He then makes mention of the fact that in many installations the liquid is bled off at different heights and that this type uses less than one-half the coil surface of a standard condenser.

If the high efficiency of the first type is due to the condensation of the vapor while immersed in the liquid, why not reverse the flow of a double-pipe condenser and increase the efficiency, and at the same time retain the counterflow feature? Also, why draw off the liquid at different levels if its presence is helpful? It would be interesting to know how Mr. Anderson was able to determine that the hot gas became submerged in a body of liquid.

Some time ago refrigerating engineers made a test of a flooded condenser that embodies the same idea of submerging the gas in a body of liquid by means of an injector. It was found that with the "submerging" coil and injector cut out, the condenser efficiency was not decreased.

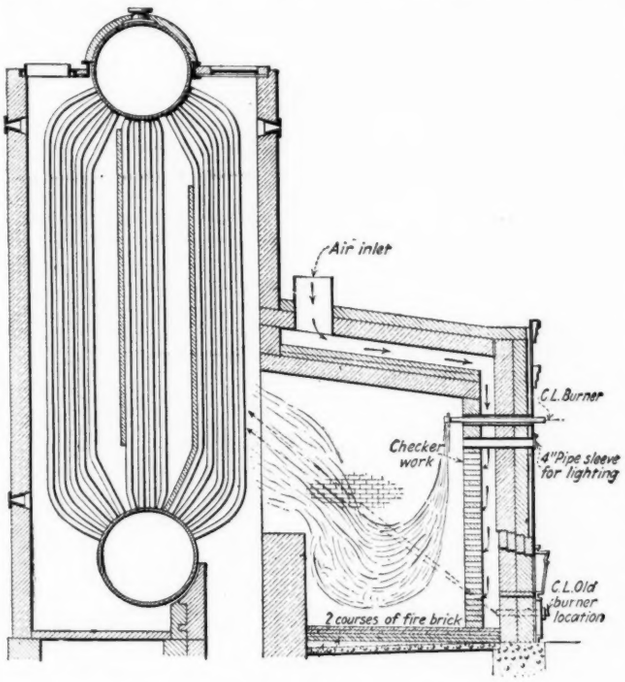
There is a second point in refrigeration practice that probably some *Power* reader can clear up. The manufacturers claim that the high pressure should be on the stuffing-box side of the expansion valve. This seems to cause more leaving than when the valve is turned around. Which is the better practice? M. H. MILLS.

St. Louis, Mo.

What Causes Tubes To Blister with Oil Burning?

Referring to the article "What Causes Tubes To Blister with Oil Burning" in the March 27 issue, I have given considerable thought to this subject and believe that the illustration shown herewith will help Mr. Hill in overcoming his trouble.

Assuming that the tube blistering in the boilers is not caused by scale-forming elements in the feed water,



DESIGN OF FURNACE PROPOSED TO ELIMINATE TUBE BLISTERING

I am led to believe that the direction of flame from the present installation would be as shown by the dotted arrows.

As there are no checkerwork slots under the burner in Mr. Hills illustration, I assume that the air is taken around the burner and with this high-velocity air entering the furnace, a stratified zone of oil and air is carried across the furnace and produces a blowpipe-like action against the tubes, probably causing overheating and blistering.

To offset this effect in steam atomizing burners in general, the mixture of the particles of air with the necessary air for combustion takes place mostly by diffusion. In order to have the necessary time for this diffusion and the resultant combustion to be completed before impingement on any cool heating surface, it is necessary to have a long flame travel.

In the present instance I believe this can best be accomplished by installing a so-called down-shot furnace.

The checkerwork arrangement for admission of air is shown in the illustration as a false wall just inside the front wall, all the air for combustion being taken in at the top of the arch, as shown by the arrows. The 4-in. pipe sleeve shown below the burner is for lighting purposes.

With the furnace constructed as shown, the flame would have a travel of approximately 18 to 20 ft., and it is doubtful even at rather high ratings whether the flame would come in contact with the tubes.

The checkerwork arrangement shown obviously has a direct relation to the draft in the furnace. It is usual practice to install this checkerwork with approximately 4.5 sq.in. per 100 lb. of water evaporated, and it is distributed in such a manner that the air will not bypass the flame. With 4.5 sq.in. per 100 lb. of evaporation, the actual draft of 100 per cent rating would probably be 0.04 inch.

This furnace was invented by Milton A. Fesler, of California, and is in my opinion a direct step forward in the way of burning fuel oil.

It is possible that increasing the height of the Dutch oven and possibly shortening it a little would give a larger area for radiating heat to the first bank of tubes. Whether this type of construction would be of much benefit would, of course, have to be the subject of some experiment. FRANK H. JOHNSON.

Boston, Mass.

The trouble experienced by Mr. Hill with blistering of tubes is not unusual with oil-burning plants. On one plant with six 6,000-sq.ft. straight-tube boilers which were converted from coal to oil, tube losses occurred at the rate of 125 tubes a month for several months. There were periods when the rate was as high as 150 a month. Even at that it was impossible to carry over 125 per cent rating on the boilers. At another plant containing eight 8,230-sq.ft. Stirling type boilers which were similarly converted from coal to oil, high tube losses occurred at high boiler rating, but not so bad as at the first plant.

The use of oil undoubtedly intensifies tube trouble owing to the higher furnace temperature and the impingement of the flame on the tubes, but it has been my experience that the oil burning is not the real source of the trouble. It can be overcome to some extent by re-arrangement of the furnaces and perhaps change of burners to prevent the impingement on the tubes. In my opinion the real source of such trouble is in the boiler-feed water. Blistered tubes are almost invariably due to scale or oil in the boiler. The use of oil for fuel simply intensifies the trouble.

At the first plant mentioned a hot-process water-softening equipment was finally installed and entirely solved the tube trouble as, after it had been put in operation and the boilers once cleaned out, no tubes were lost for a period of six months. At the second plant internal treatment of the feed water with soda ash is being tried out. This has not been in use long enough to give definite results, but it has already made a decided reduction in the tube renewals even with operation at higher rating. D. F. EVANS.

Wilmington, Del.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Forcing a Boiler

What is meant by "forcing" a boiler? It has been claimed that boilers can be operated at 300 per cent rating or more and not be forced. H. D. N.

The engineering significance of the word forcing, as applied to boilers, refers to the realized rate of heat transfer to the water and not to the degree of urgency of the firing. The normal rate generally is assumed to be equivalent to the heat required for evaporation of 3.45 lb. of water from and at 212 deg. F. for each square foot of heating surface, per hour. When that rate is exceeded, the boiler's capacity is said to be forced beyond its rating. When double the normal rate per square foot of heating surface is realized, the boiler would be operated at 200 per cent of rating and thereby forced 100 per cent beyond rating.

Trouble from Condensate Freezing in Hoisting Engine

We have a piston-valve hoisting engine that is located in a pit about ten feet below the main steam line. A trap relieves condensation that may accumulate in a receiver that is in the steam line near the engine; but there is not provision for draining the drop of the line to the engine, and during cold weather freezing of condensate resulted in bursting the cylinder of the engine. How can the accumulation of water be prevented when the engine is not in use? W. H.

There should be a trapped separator in the steam line as close as possible to the engine. Undoubtedly, the engine steam chest and cylinder are provided with pet cocks for drainage of water, and these should be left open a little at all times when the hoisting engine is not running.

Hydrostatic Test Pressure for Boiler

How much test pressure should a boiler be subjected to, in excess of the allowable working pressure, and what objection if any is there to a cold-water pressure test? A. F. B.

When a hydrostatic test is applied, the required test pressure usually is one and one-half times the maximum allowable working pressure, with the pressure under proper control so that the required test pressure will not be exceeded by more than 2 per cent. The allowable working pressure is to be determined from competent inspection and approved methods of computation of strength of parts, and application of a factor of safety compatible with the workmanship, condition and length of service of the boiler. The hydrostatic test pressure is not to be regarded as the safe working pressure, but only as an assurance that the pressure-resisting surfaces, stays, fastening and joints will remain intact when the boiler is operated at the allowable working pressure.

Objection sometimes is made to subjecting a boiler to so much more than the allowable working pressure because the material is stressed beyond the stresses induced by the working pressure, and is likely to assume a set, causing stays to be slack when the boiler is operated at the working pressure; also that this tendency is greater from applying the test pressure with cold water, from contraction in length of the stays while subjected to test under lower temperature. However, both of these objections are reasonably met in modern boiler construction by approved character of workmanship and material and employment of factors of safety that are indicated by experience to be adequate.

Trouble from Straight Tube Heater

We have a closed exhaust-steam feed-water heater, built like a return-tubular boiler, with straight tubes. The tubes are brass and cannot be kept tight in the tube sheets. How can the trouble be prevented? Would not spiral form of tubes be better? W. H. C.

Straight tube heaters generally give trouble from leakage at joints of the tubes in the tube sheets, especially where the tubes and shell are of different kinds of material. There should be flexibility in the tubes to allow variation in the distance apart of the fastenings of opposite ends, such as from having the tubes in the form of coils; or each tube should be made of U form to receive the exhaust into one leg and discharge it out of the other. With straight tubes there is no compensation for difference of expansion of the tubes and of the shell, and with each change of temperature there is disturbance of the joints, inducing leakage.

It would be impractical to provide slip joints for one end of the tube in order to continue the use of the present heater as a closed heater between the boiler-feed pump and the boilers. However, it might be used as an upright open heater employing the tubes as heating surfaces, by the admission of exhaust at the lower ends of the tubes, with the tubes slipping freely through the upper tube sheet and made tight in the lower tube sheet, but with ample drips to prevent drainage of any leakage back to the exhaust connections of the engine. Such an arrangement would require the upper end of the shell to be vented to the atmosphere, and there should be a glass gage for indicating the level of water contained by the shell and also an overflow near the top of the shell. The cold-water supply should be received at the top or through a connection at the bottom of the shell extended to the top of the interior, with discharge to the boiler-feed pump suction taken out of the bottom and the pump placed several feet below the heater.

Form of Pump-Valve Springs

What are the relative advantages of pump-valve springs made in the form of a cylindrical helix and when made in the form of a conical helix? R. C.

When valve springs are made in form of a conical helix, as shown in Fig. 1, there is an advantage in requiring a simpler form of spindle for holding the spring central with the valve, and the conical form also has the advantage over the cylindrical helix in permitting the valve to rise nearly the full height of the unloaded spring to the form of a spiral without the coils coming together. But the conical form is defective in that the upper coils are stiffer than the others and, from being held more firmly, breakage frequently occurs in those coils. When a spring is made in the cylindrical form shown in Fig. 2, there is more uniform stress throughout the coils and more gradual change in bearing of the upper coil when the spring is compressed in length from opening of the valve, and consequently greater durability of the material.

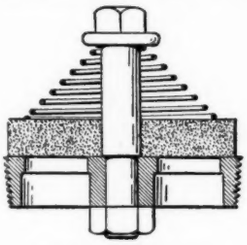


Fig. 1—Conical form

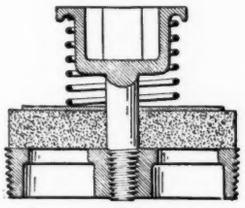


Fig. 2—Cylindrical form

PUMP VALVE SPRINGS

Operating Motor-Generator Set Inverted

In our plant there is a 150-kw. motor-generator set, consisting of a 240-volt direct-current generator driven by a 440-volt three-phase 60-cycle induction motor. Although the set was primarily intended to transfer power from the alternating-current to the direct-current system and has always been used for this purpose, there are times when it would be advantageous to supply power from the direct to the alternating-current system. Could the set be used for this purpose, and what changes are necessary? A. J. M.

If the direct-current machine is shunt wound, no changes are required in the connections to operate the generator as a direct-current motor and the induction motor as an induction generator. All that is necessary is to reduce the voltage by cutting in the field rheostat until the generator load decreases to zero and the machine is operating as a motor. The amount of load taken by the induction generator will depend on its speed; to carry full load as a generator, it will have to operate at about the same percentage of speed above synchronism as it operated at below synchronism when running as a motor under full load. In other words, if at full load and operating as a motor the slip was 5 per cent, then as a generator to carry full load the speed will have to be about 5 per cent above synchronous speed.

If the generator is compound-wound, then some means will have to be provided for crossing the series field coil connection when the set is operated inverted, or the series winding will have to be short-circuited with a single-pole switch. The latter will probably be the most satisfactory, since in changing from generator to motor the voltage can be reduced until there is only a light load on the machine and the single-pole switch closed, which should cause the generator to drop its load entirely and begin operation as a motor. After this the operation is the same as for a shunt machine. In going back into regular service, the field of the direct-current machine is increased until it is carrying a small load as a generator; then the single-pole switch is opened. The single-pole switch will have to be located near the machine and the connections made of substantial proportions, to maintain the resistance of the switch circuit low enough to be effective in short-circuiting the series-field winding.

When the set is inverted, the wattmeters on both ends of the set and the direct-current ammeter will be reversed. The set will not have as high a rating when operating inverted as when converting alternating into direct current, since in the former the direct-current machine will supply not only its own losses, but also those of the alternating-current machine. It is to be understood that the induction motor can be converted into a generator only when it is connected to some source of alternating-current supply.

Atmospheric Pressure and Suction Lift

What is the true atmospheric pressure against a perfect vacuum at sea level, and how high will water be raised in a suction pipe in which the pressure is 10 lb. per sq.in. below the pressure of the atmosphere? C. J. K.

The pressure of the atmosphere at sea level is slightly different at different latitudes for the same conditions of atmosphere, and varies at any particular place with changes of weather. The standard pressure of the atmosphere assumed by physicists and engineers is the intensity of pressure that would be exerted by a column of mercury 760 millimeters = 29.921 in. high, at the standard temperature of 32 deg. F., which is equivalent to 14.696 lb. per sq.in. or, as usually taken, 14.7 lb. per sq.in. The true atmospheric pressure at sea level at any time is shown by the height of a barometer after making due allowance for the temperature of the mercury column.

When the pressure in the upper end of a suction pipe is 10 lb. per sq.in. less than the pressure of the atmosphere, the atmosphere would balance a still column of water of such height that the weight of the water would exert a pressure of 10 lb. per sq.in. When the water is at the temperature of 62 deg. F., for each foot of height there would be a pressure of 0.433 lb. per sq.in. and consequently the height of a quiet water column sustained above the level of the tank or reservoir would be $10 \div 0.433 = 23.09$ ft., which is equal to 23 ft. and about 1 in.

However, if the column of water is in motion, some of the pressure exerted by the atmosphere would be lost in overcoming frictional resistances to the flow at the entrance and friction of the suction pipe and fittings. Hence the height to which the water could be continuously raised by a pump with a pressure in the upper end of the suction pipe of 10 lb. per sq.in. less than the pressure of the atmosphere, would depend on how much pressure would be required for overcoming suction pipe friction.

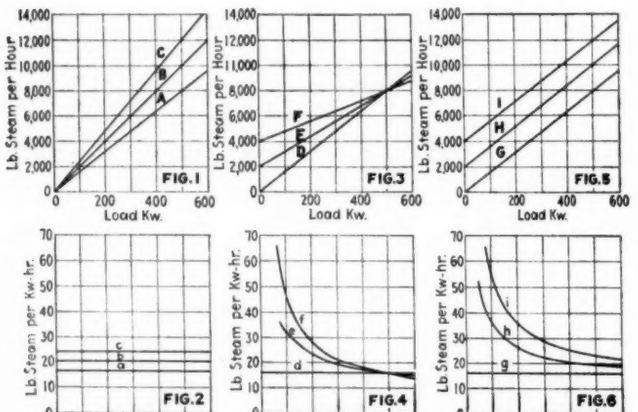
[Correspondents sending us inquiries should sign their communications with full names and post office addresses.—Editor.]

Easy Lessons in Engineering

The Meaning of Steam-Consumption Curves

As was stated in the article "Plotting a Turbine Test," in the previous issue, plotted curves of engineering data have a wide field of usefulness. One important practical application—the plotting of steam-consumption data during the course of a test—was explained in that article. Countless other examples could be given if space permitted. What will be done in the present article is to consider the meaning of the shape of curves, taking as examples various curves of steam consumption.

In the previous article it was explained that (at least for a throttle-governed turbine or engine) the



FIGS. 1 TO 6—STEAM-CONSUMPTION CURVES

curve obtained by plotting the total steam consumption against the kilowatt output was practically a straight line (called the Willans line). It was also shown how a reliable curve of steam consumption per kilowatt-hour could be plotted from data taken from the Willans line.

We will now consider what effect the position of this line has on the shape and position of the corresponding steam-rate line. Although such a condition is impossible in practice, it will help us to understand the problem if we start with Willans lines of imaginary units whose steam consumption at no load is zero. Three such lines (*A*, *B* and *C*) are shown in Fig. 1, and their corresponding steam-rate curves (*a*, *b* and *c*) in Fig. 2.

First considering *A* of Fig. 1, it will be noted that the hourly steam consumption is in exact proportion to the load—16 times as much as the load at all points. This proportional relation is true of any straight-line graph that starts at the zero of both scales, for the height of any point on such a line is evidently in exact proportion to its horizontal distance from the starting point. To plot the steam-rate curve *a* from *A*, various loads are taken at random and divided into the corresponding total steam consumptions. This gives the steam rate, which is found to be 16 lb. per kw.-hr. for every point on *A* and is so plotted in *a*.

Similarly, if it were possible to obtain Willans lines like *B* and *C*, of Fig. 1, the corresponding steam-rate curves would be as shown at *b* and *c* in Fig. 2. In other words, any Willans line starting at zero on both scales will give a horizontal steam-rate line at a height equal to the slope (or rate of increase) of the Willans line.

This does not hold for actual Willans lines, which always show a considerable steam consumption at no load. Such lines are shown at *E* and *F* in Fig. 3. These, together with the line *D* (which is put in for comparison and is the same as *A* of Fig. 1), all intersect at 500 kw., where they have the same steam consumption of 8,000 lb. per hour. When the corresponding steam-rate curves (*d*, *e* and *f* of Fig. 4) are plotted, they must also coincide (or intersect) at 500 kw. with a steam rate of 16 lb. per kw.-hr. For all loads below 500 kw. *e* is greater than *d* and *f* is greater than *e*. In other words (for a given steam rate at rated load) the smaller the total steam consumption at no load the "flatter" will be the steam-rate curve. If the unit were to be operated continuously at 500 kw., it would make no difference what the no-load total steam consumption happened to be. When, on the other hand, the unit must operate much of the time at fractional loads, the best economy is obtained where the no-load total steam consumption is a minimum or, in other words, where the steam-rate curve is as flat as possible.

A third kind of variation is illustrated by Figs. 5 and 6. The Willans lines *G*, *H* and *I* in Fig. 5 all have the same slope. Moreover, *G* is exactly the same as *A* of Fig. 1, and *D* of Fig. 3, and increases at the rate of 16 lb. of steam for each kilowatt increase in load. With *H* and *I* the rate of increase is the same, but the actual total steam consumption is greater by the amount of the steam consumption at no load. When the corresponding steam rates are plotted in Fig. 6, it is found that *h* exceeds *g* and *i* exceeds *h* at all points, but that the amount of the excess decreases as the load goes up. It should be noted that the curves of Figs. 5 and 6 correspond with those of Figs. 1 and 2 at the 500-kw. point.

From a study of these diagrams the following conclusions may be drawn: (1) Lowering the no-load total steam consumption flattens the steam-rate curve with a resulting increase in economy at fractional loads; (2) decreasing the total steam consumption the same amount at no load and full load decreases the steam rate at all loads, but least at the greatest loads and most at the smallest loads.

The Cross-Flow Impulse Hydraulic Turbine*

A New Type of Impulse Turbine Designed on Principles That Permit Much Higher Specific Speeds Than Are Possible with the Usual Design

BY FORREST NAGLER†

CONTRARY to the general impression the modern impulse wheel is the slowest-speed type of hydraulic turbine known, although it utilizes the highest water velocities. This low-speed type has found an extensive field of usefulness by reason of its mechanical simplicity and the relatively small surface exposed to high water velocities. Its mechanical simplicity permitted the original development of this type without going beyond the capacity

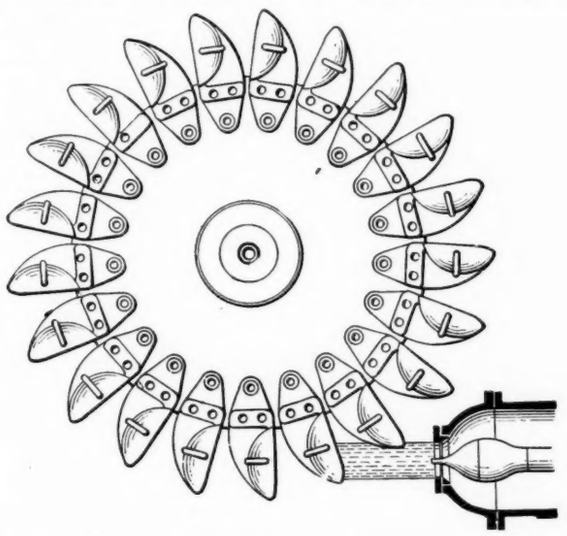


FIG. 1—HOW JET IS APPLIED TO TANGENTIAL TYPE OF IMPULSE TURBINE

of carpenter's and blacksmith's tools. Its low speed was essential in connection with small capacities and high heads to which it was adapted, as this permitted speeds sufficiently low to allow readily of direct connection to alternators of reasonable speeds. Its arrangement permits elimination of clearances and packing, which involve careful design and fine machine work and are a continual source of trouble under high heads. The small amount of surface exposed to water flow and the ease with which such surfaces may be inspected and renewed overcame one of the greatest problems in high-head turbine design.

So far as water velocity is concerned, the impulse wheel should have the highest speed of any type of turbine, except those which may be classed within the so-called "suction" field. The water velocity in the usual reaction wheel (Francis) seldom exceeds 50 to 75 per cent of the spouting velocity, or the same percentage that prevails in impulse practice. The main reason for the low characteristic speed of impulse wheels is due to using the whole periphery of the Francis type for developing powers, whereas the usual impulse wheel comes under the classification of partial turbine. The result is that in comparing on the basis of a certain horsepower the diameter of the impulse type becomes so large that its speed is unduly reduced. This is further emphasized by the fact that the reaction wheel usually runs with a coefficient of rim velocity in the neighborhood of 60 to 90 per cent as contrasted to the 50 per cent for the present impulse type.

Speeds of tangential impulse wheels cannot be increased by reducing their diameter beyond a certain limit because the bucket turns so abruptly out of the working path of the jet that there will be what is called "racing water," with its inherent loss of efficiency.

Unit capacities have so grown that for a considerable period efforts have been made to increase specific speeds of impulse units. As a result the art of impulse-wheel design went through the same evolution as did that of reaction-type turbines; that is, multiplicity of runners, nozzles, etc., without attacking the fundamental element of relative bucket velocity or direction of jets on the wheel to avoid interference.

EXPERIMENTAL WORK ON TURBINES

During the years 1913 to 1918 the author was experimenting with flat-angle runners of the axial-flow type. This work resulted in the development of the high-speed suction-type propeller runner since applied extensively to low heads. This led to an appreciation of the characteristics necessary to high speed in any vane moving in a fluid. In a paper describing this work before the A.S.M.E. in December, 1919, the following statement was made as covering any waterwheel, the ice-boat illustration incidentally coming under the classification "impulse" as defined herein.

Neglecting friction and possibly blade thickness, there are no mathematical or hydraulic laws that will prevent doubling or quadrupling any particular characteristic speed by simply flattening the blade angles. A direct analogy to this is the well-known illustration of relative velocities evidences in the sail of an ice boat.

Continuing this work on high-speed runners with impulse wheels, the author, in company with J. F. Roberts, experimented with small-angle single-lobe types of buckets with the jet making small angles with the tangent. The purpose was to secure an arrangement whereby more jets might be used on a single wheel without interference and to eliminate the splash losses resulting from water discharged upward and falling back on the wheel in vertical shaft arrangement. In playing jets from an ordinary

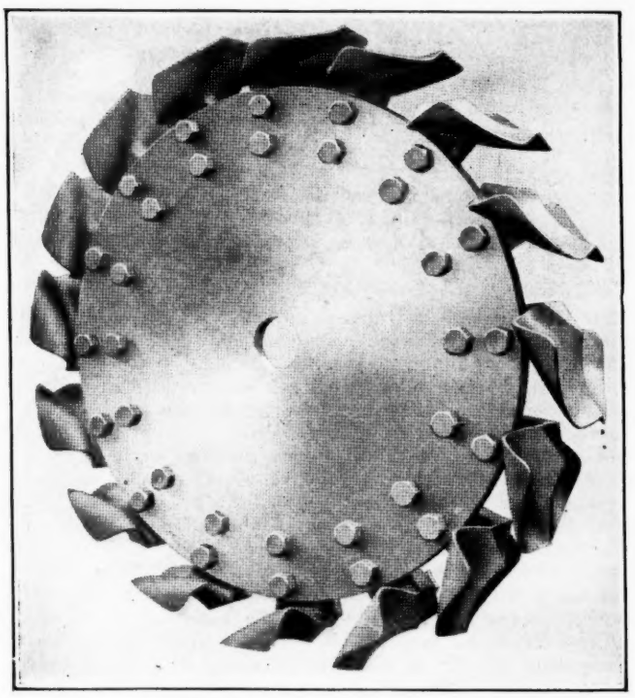


FIG. 2—ASSEMBLY OF CROSS-FLOW TYPE RUNNER

*Abstract of paper presented at the Pacific Coast meeting of the American Society of Mechanical Engineers, April 16 to 18, 1923.

†Hydraulic Engineer, Allis-Chalmers Manufacturing Co.

garden hose on small models of these wheels that were running at a high speed (10,000 to 20,000 r.p.m.), a decided change in their tune was noticed with various positions of the jet. On account of ease in construction one of the wheels happened to have been cast without back curvature of buckets, similar to the propeller-type suction runner. With this wheel the highest pitch or note was developed with the jet directed almost perpendicular to the plane of the wheel. This was so contrary to fundamental impulse practice as indicated by all commercial installations built in recent years and covered in any treatise on hydraulics, that it attracted instant attention. Analysis of the condition resulted in a design called the cross-flow type. Instead of the water's striking the buckets at a tangent to the rim of the wheel Fig. 1, it is applied at an angle greater than 45 deg. to the tangent, as in Fig. 3, with buckets shaped similar to those shown at *A*. Fig. 2 shows the assembly of a runner of cross-flow type.

Speed in impulse-wheel work is a function of the angle the jet makes with a tangent. If the jet angle is zero, as in accepted present practice, the speed is the lowest that can be obtained without preventable loss. As the angle of the jet increases to 45 deg., the relative wheel velocity increases from 50 per cent to 69 per cent of the jet velocity. From this point, however, the most radical increase is effected. For example, from a 69 per cent coefficient at 45 deg. the speed increases to 100 per cent at 61 deg. This feature, combined with the obvious departure from the tangential flow of present practice, led to the following designation of two classes of impulse wheels accordingly as the major component of the jet velocity is along or normal to the tangent:

Tangential impulse wheels—Jets making angle less than 45 deg. with tangent; buckets decidedly concave, curvature greater than 90 deg.; bucket inlet inclined backward.

Cross-flow impulse wheels—Jets making angle greater than 45 deg. with tangent; buckets flattened, curvature less than 90 deg.; bucket inlet inclined forward.

The name cross-flow seemed especially appropriate not only because it so aptly describes the direction of flow across the wheel, but because it so distinctly defines the direction of flow as contrasted to that in the present wheels, which are universally known as the tangential type.

The cross-flow wheel, aside from its advantage in speed, automatically corrects one of the defects that contributed greatly to the failure of single-flow impulse turbines such as the Girard and other partial wheels. The backing of water in the wheel results in its being dragged around and ultimately discharged at wheel velocity at about half its original velocity. The slower the speed of the bucket the greater the curvature and consequently the greater the impact and back-water loss. With higher speeds the bucket becomes flatter and the backing loss with its poorer efficiency and greater tendency to pit is more and more reduced without resorting to the undesirable expedient of increasing the number of buckets. The author believes that this principle can be utilized to eliminate or at least reduce the efficiency losses and pitting troubles experienced with the Girard types, and with it there will be a rapid return to radial and axial and even conical or mixed-flow impulse wheels. In combination with nozzles designed to deliver circular jets with their higher efficiencies and more uniform distribution of velocities cross-flow wheels can be applied to a considerable portion of the field, for which there is at present no design except the disadvantageous multiple-runner or multiple-nozzle types. For low heads

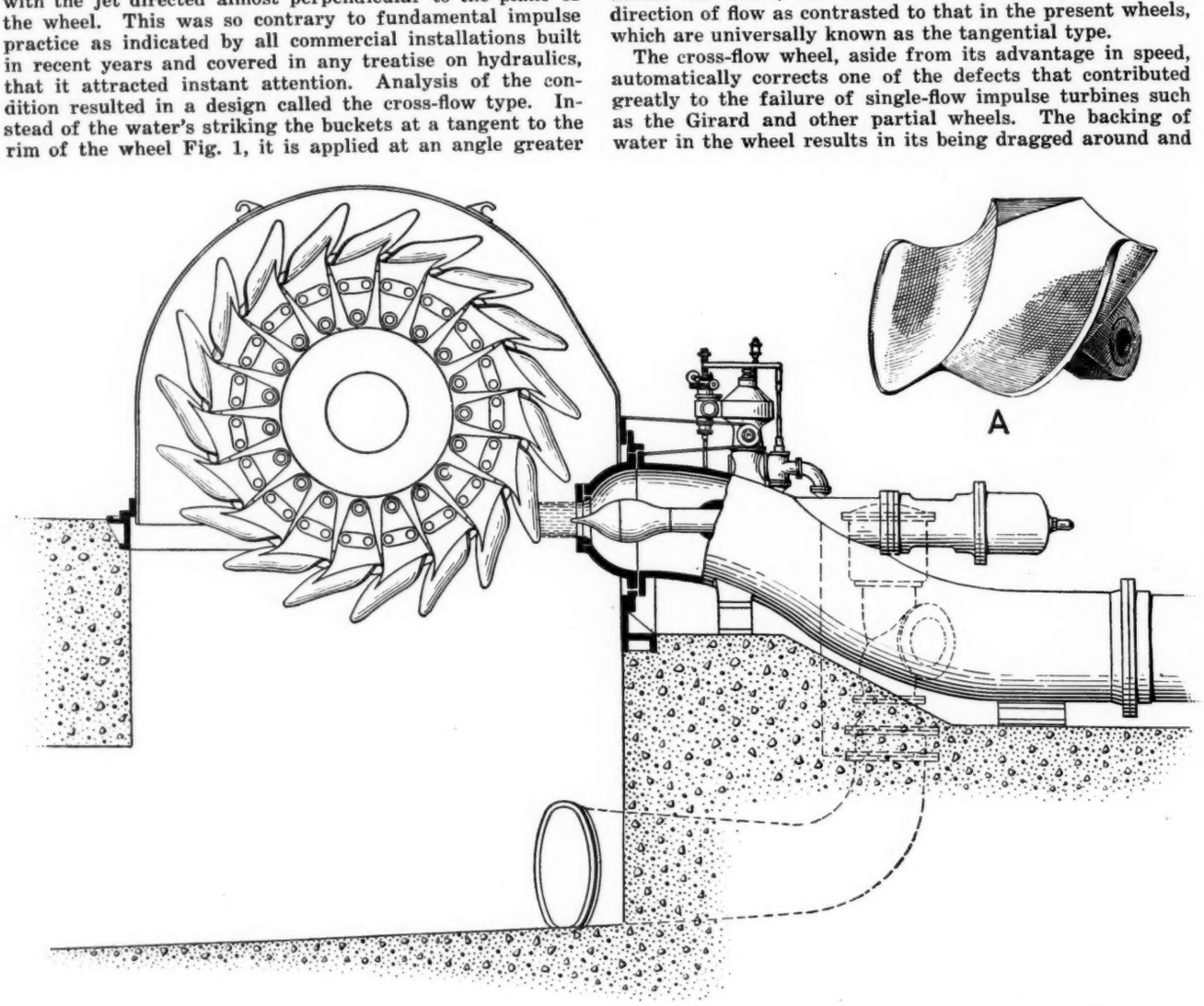


FIG. 3—SUGGESTED APPLICATION OF THE CROSS-FLOW PRINCIPLE TO AN IMPULSE-WHEEL UNIT OF CONVENTIONAL DESIGN

The large angle of incidence shown is obtained by simply raising the jet above a tangential line so that it flows in a more nearly radial direction. *A* shows a bucket for use with a cross-flow wheel.

and small powers or where efficiency may not be of the greatest importance the circular jet may be dispensed with and rectangular jets arranged to cover partly the wheel periphery (partial turbines), or even solid annular jets (full turbines) may be utilized.

The cross-flow wheel does have an inherent disadvantage from an efficiency viewpoint. The relative velocity between jet and bucket is the lowest (50 per cent of spouting velocity) in a tangential wheel and higher in the cross-flow types; for example, for a 100 per cent coefficient the relative velocity is practically twice what it is in a tangential wheel. This, of course, involves a greater friction loss, but in actual practice this may be offset by reduction of windage and splash losses and particularly by the possibility of developing more power in a given working volume of space swept by the buckets than it is possible to do with the tangential type.

Advantages of Vertical Impulse Wheels

Vertical impulse wheels have certain mechanical advantages when large capacities are considered. There are similar to those which underlie the popularity and economies of the vertical reaction units and in addition incorporate the possibility of feasible mechanical arrangement of more jets than possible with horizontal types. Engineers experienced with impulse work, however, are familiar with the tremendous disturbances that exist even in horizontal wheels having but a single jet, and can appreciate the condition that must exist when several jets are played on one wheel. When it is further considered that all the upwardly discharged water that falls or is spattered back into the path of the buckets must again leave the wheel at practically wheel velocity, or half its own initial velocity, and that this represents a total loss, the desirability of getting the water positively and clearly away from the wheel is apparent. The cross-flow design, embodying as it does the feature of single direction of flow through the wheel without backed-water losses, lends itself to this arrangement. This is particularly so as in extreme cases multiple or annular jets may be used without injurious interference, as each jet uses a relatively small portion of the periphery as contrasted to the tangential type, where each jet needs for its working space a chord subtending a relatively larger angle.

There is a comparatively undeveloped gap existing between the impulse and reaction types. This gap exists by reason of the difference between the highest impulse characteristic speed (about 5 ft.-lb. units) and the lowest reaction characteristic speed (about 10). Even the multiplication of jets and nozzles incorporated in most of the larger-capacity impulse units has failed to bring the two fields together. The field of head and capacity now covered by impulse wheels of the tangential type needs extending only a limited amount to cover the section between the fields to which tangential impulse and reaction wheels are well adapted. Going beyond this neutral ground in the direction of higher specific speeds involves competition with the reaction-type runners that give economical generator speeds with lower water velocities and better efficiences. It is to this intermediate field that the cross-flow impulse design is peculiarly adapted, since with it characteristic speeds of 5 to 20 are readily obtainable with single jets. With multiple or annular jets even higher speeds are possible, theoretically up to over 100. There will probably be no commercial demand for such extreme speeds using the cross-flow principle, as the reaction type covers this field satisfactorily and with more desirable water velocities. The greater simplicity of the impulse type may, however, make those high-speed types work out advantageously with small or auxiliary units.

Why New Designs Are Needed

It is for lower heads and larger capacities that new designs are needed. While the elements of the machines themselves are not subjected to mechanical complication or hydraulic losses as serious as those which caused the multiple-runner reaction turbine to be superseded by the single vertical arrangement, the general complications of water passages and governing mechanisms and possibilities of flow interference are certainly such as to be expensive and undesirable. If the history of the progress in reaction wheels is any guide, future developments in the impulse field will return to single runners of higher characteristic speeds, just as the multiple-runner reaction wheels yielded first to higher-speed mixed-flow wheels and later to the still higher-speed axial-flow or suction wheels.

Fig. 3 is shown as an example of proportions and possible details of a unit based on the cross-flow principle for conditions of reduced head and increased capacity comparable with those that incur multiple jets or wheels. This is shown as indicating a possible trend in impulse design based on investigation of modern commercial needs and analysis of the hydraulic and mechanical features involved. It is not suggested that initial installations should be made for such large capacities and heads (15,000 hp., 1,008 ft., 360 r.p.m.) without first obtaining more operating data than are now available on the new type proposed. It is believed, however, that the low-head extension of the field of impulse wheels, particularly for small and moderate capacities, is warranted immediately.

The main purpose of this paper is not to present a complete solution of problems confronting the designer of impulse wheels, nor is it intended even to present the new type as being completely worked out. It is intended to draw attention to the possibility of departing from the orthodox tangential flow and 50 per cent coefficient, departing even from the specter of practice, precedent, usage and textbooks without violating perfectly sound hydraulics. If the only too common tendency to get in a rut and stay there is lessened, the author will be more than satisfied.

Europe Is Developing More New Water Power Than the United States

A summary of the estimated potential and developed water power in the world in 1920, by countries, was published by the Geological Survey in 1921 in a "World Atlas of Commercial Geology, Part II: Water Power of the World." Progress in Europe in the two years since that report was issued is shown by reports from United States consular officers. The following notes, arranged by countries, have been taken mostly from consular reports.

Austria—The developed water power in 1920 amounted to 170,000 hp. This will be nearly doubled by construction started in 1921 and 1922 on plants that will have a capacity of 162,000 horsepower.

Czechoslovakia—Two hydro-electric plants are now under construction in Bohemia.

Finland—The total capacity of water-power plants is now over 200,000 hp. The capacity of the Voikka plant, on Kymmene River 75 miles northeast of Helsingfors, has been increased from 6,500 to 20,000 hp. Construction work is in progress at Veroken Rapids, at Imatra, on a plant whose turbine capacity will be 70,000 hp. At this place 680,000 hp. can ultimately be obtained by controlling Saima Lake, the area of which is 655 square miles, or about one-twelfth that of Lake Ontario.

France—An increase of 50 per cent in two years in the capacity of waterplants gives the capacity in 1922 as 2,100,000 hp. This total probably includes plants under construction. The largest projects proposed are on the Rhone, 900,000 hp.; on the Dordogne, 200,000; on the Truyere, 200,000.

Germany—In Bavaria the water-power plants already constructed have a capacity of 347,000 hp., and those under construction an ultimate capacity of 457,000 hp. The largest of the new plants, within 60 miles of Munich, are at Walchen See on the Isar, ultimate capacity 144,000 hp.; on the middle Isar, 110,000; and on Inn River at Jettenbach, 100,000. A proposed canal to connect the Rhine and the Danube would develop 400,000 horsepower.

Greece—The developed water power amounts to 6,880 hp., to which should be added 1,000 hp. in plants practically completed to supply two modern spinning mills.

Hungary—The developed water power in the present area of Hungary amounts to only 3,400 hp. Projected plants would have a capacity of 34,000 horsepower.

Italy—In 1898 the capacity of electric plants, including both water-power and steam plants, was 116,000 hp., in 1908 it had increased to 568,000, and in 1918 to 1,650,000 hp. The greater part of this power is generated by water. The largest hydro-electric project under construction is the Piave and Lake Santa Croce project, which is practically completed to 50,000 hp. and will have an ultimate capacity of 200,000 hp. Two projects under construction in Sicily will have a capacity of 27,000 horsepower.

Jugoslavia — The developed waterpower amounts to 160,000 horsepower.

Norway—The Solberg plant, on Glommen River, with a capacity of 70,000 hp., will be completed in 1923. A very large state-owned project is now under construction at the outlet of Lake Tunhovd on Laagen River 75 miles northwest of Kristiania.

Portugal—A plant of 10,000 hp. has been built on the Spanish-Portuguese frontier of ultimate capacity of 40,000 horsepower.

Spain—The Cinca Falls plant, of 40,000 hp., which will supply power to Bilbao, is about completed. A plant of 38,000 hp. has been built at the junction of Juca and Gabriel Rivers. Plants under construction will supply Madrid with additional power amounting to 6,000,000 kilowatt-hours.

Sweden—The capacity of water-power plants under construction in 1921 amounted to 360,000 hp., of which 235,000 was in State projects.

Switzerland—The water-power plants in 1920 numbered 6,870, and the total maximum capacity was 1,362,000 hp. The Federal Railways have recently completed a plant at Amsteg on the Rouss, 68,000 hp.; and one at Ritom near Amstog, 52,000 hp. installed and 78,000 ultimate capacity. These two plants supply power to the Gotthard Railway and tunnel. The Federal Railways are also building a plant at Barbarine Falls in Valais, with an initial capacity of 41,000 hp., to be completed in 1924; and the Aarau-Wildeg station on the Aar, with a maximum capacity of 60,000 hp., to be completed in 1923. Another large plant has recently been completed at Muhleberg on the Aar, with 49,000 hp. Other stations under construction are the Broc, on the Jogne, Motelon, and Javrez Rivers, 20 miles northeast of Lake Geneva, with a capacity of 24,000 hp.; the Klosters-Kublis, on the Landquart, a headwater tributary of the Rhine, with an initial capacity of 20,000 hp. and an ultimate capacity of 55,000; the Chancy-Pougny, on the Rhone, with a capacity of 43,000 hp.; the Davos-Klosters, near Davos, with a capacity of 30,000 hp.; one at Siebnen, 25 miles southeast of Zurich, 60,000 hp.; and one at Rampen in the canton of Schwytz, 80,000 horsepower.

Student Engineers Hold Meeting

The Eighth Annual Intercollegiate Convention of the Student Branches of the American Society of Mechanical Engineers, located at the several engineering colleges in and near New York City, was held on Tuesday, April 27, at the Engineering Societies Building, New York City.

Papers on European boiler plants, Japanese engineering progress, oxyacetylene welding and the Hudson River Vehicular Tunnel were presented.

From a mechanical-engineering viewpoint probably the most interesting of the papers was "Random Notes on European Boiler Plants," by Col. Frederick Pope, vice-president, New York Steam Corporation. Colonel Pope has recently returned from an inspection of prominent plants both in England and on the Continent. He found that in boiler design the United States is far ahead of Europe, especially in respect to capacity. As an example, in German plants the boilers are of comparatively small capacity; often ten are used where we would use a single boiler. Also, the boilers are not operated at the high overating as in this country, an evaporation of five pounds per square foot of heating surface being the usual practice.

In furnace design a similar contrast in practice confronts one. The European engineer is content with furnace volume to heating-surface ratios far below those demanded by American engineers, even taking into consideration the higher evaporation in American boilers.. That the European furnace does not suffer in efficiency as a result of its small volume is evidenced by the results obtained in burning low-grade fuels. Mr. Pope showed that with chain-grate stoker an efficiency well over 70 per cent was obtained from lignite, high-volatile bituminous and from a low-volatile hard coal; in fact, efficiencies of over 80 per cent were not exceptional.

In ash handling the European practice is to use some form of chain buckets, and it is seldom that steam ejectors or pneumatic systems are used. In one plant cited, the entire system of chain-link buckets, etc., was made of tool steel with the idea of reducing wear.

A more extensive use of copper expansion joints is found in English plants. The practice there is to eliminate all gaskets; the flanges are faced, one being turned with a series of small ridges, and in drawing up the bolts these ridges are sunk into the face of the companion flange. In a number of the larger plants the flanges are also welded.

Air heaters are becoming a standard adjunct of the European boiler plant, and Mr. Pope predicts that American plants will soon adopt this means of heat recovery; it had been found that the saving is about 2½ to 3 per cent of the total heat in the fuel. Mr. Pope illustrated his lecture with a number of lantern slides showing typical plants.

Following Mr. Pope, Elmer A. Sperry gave a brief talk on his visit to Japan.

Largest Waterwheel Generator Ever Made

This unit has been ordered by the Niagara Falls Power Company for its No. 3 power station and is the first of two units of the same size being built by the General Electric Co. Each generator will weigh 700 tons and will be 26 ft. high and 35 ft. in diameter. Both as to size and capacity

MACHINING THE STATOR OF A 65,000-KVA. GENERATOR IN THE SHOPS OF THE GENERAL ELECTRIC CO., SCHENECTADY

they are the largest in the world. They will be driven by 70,000-hp. hydraulic turbines made by the I. P. Morris Department of William Cramp Sons Shipbuilding and Engine Co., Philadelphia.

At furnace heat the hydrocarbons in the fuel are vaporized and burn like a gas, except that the hydrogen usually burns a little ahead of the carbon. That leaves extremely fine soot-like particles of carbon which glow at red or white heat (depending on the temperature) until they are consumed. These incandescent particles make the flame. If the flame strikes the cold tubes, the unburned particles of carbon pass on as smoke; so the solution of the smoke problem is simply a question of having combustion complete before the gases enter the tubes. The visible smoke itself is mainly unburned carbon, but its presence usually shows that other combustible matter is being lost as well. The invisible loss is generally the larger.

News in the Field of Power

N. Y. Constitution May Be Amended for Water Power

The most important water-power measure introduced at Albany this winter is the constitutional amendment presented by Senator Ferris, which provides, in effect, that the legislature may provide by general laws, for the use of the forest preserves for the development of water power for the public benefit, and for the construction, maintenance and operation of ponds, structures, conduits and appurtenances necessary. Such power is not to be used, however, for manufacturing or business purposes. The development may be made by the state or by a lessee of the state under a lease for a term not exceeding fifty years. Not to exceed 3 per cent of the forest preserve may be used for the construction of transmission lines, transmission stations and dams.

Under present constitutional limitations, whether the state itself, private corporations or the United States government under the provisions of the Esch law were to develop hydro-electric energy at the St. Lawrence, it would be impossible to transmit it in a direct route to the capitol district and New York, because under the provisions of the New York State Constitution, not a stick of timber may be cut on the preserve for any purpose, and this would mean that transmission lines from Long Sault Rapids would by force of necessity be carried either by way of Plattsburg and thence southerly or via Malone, Ogdensburg, Carthage and Utica around the present Adirondack preserves.

The amendment conforms to the national Esch law in that it limits the term of lease to private corporations to fifty years. If approved by the legislature, the measure will be presented to the voters this fall, and whatever policy the state may ultimately adopt, if the amendment be approved, the State of New York will be in a position to cooperate intelligently in the transmission of any electricity generated in the St. Lawrence River watershed.

Clarion Water-Power Project Started

The Penn Public Service Corporation has just placed the first order for electric equipment for the Clarion River development with the Westinghouse Electric & Manufacturing Co. It consists of two 12,000-kva. vertical water-wheel generators with transformers and switching equipment. This equipment will be installed in the Piney Creek power house, which will be the first of the three generating stations in the project to be put in operation. It is expected that this plant will be ready in 1923.

The plans for the Clarion project call for the construction of three dams and generating plants. One of these, known as the Foxburg development, is about two miles from the mouth of the Clarion River, which empties into the Allegheny about sixty miles north of Pittsburgh. The second, the Piney development, is about 24 miles upstream. The third is still farther upstream, about six miles above the town of Clarion, Pa., and is known as the Mill Creek development. The dam at Piney will be about 75 ft. high, that at Foxburg about 175 ft., and that at Mill Creek about 250 ft. high.

The ultimate installation at Piney will be 44,000 hp., 88,000 hp. at Foxburg, and 130,000 hp. at Mill Creek. One-half of the apparatus for the Piney plant is included in this first order. There are seven 4,000-kva. transformers in the order, outdoor type, single-phase, 63,584 and 38,150 volts delta, 110,000/66,000 volts star high tension to 11,000 volts delta low tension, and there is also ordered a 50-kw. 250-volt gasoline-engine-driven exciter, consisting of a 180 SK generator set driven by a four-cylinder Sterling gas engine, which will be connected to the station auxiliary bus for emergency use.

When completed, the project will impound about 47,550,000,000 cu.ft. of water in the watershed of the Clarion River and its lower tributaries, a volume sufficient to develop 400,000 hp. of electric energy. A total expenditure of about $28,000,000 will be made to complete the project. It is proposed to use the electric power for the industrial development of northwestern and western Pennsylvania and eastern Ohio.

Connecticut Co. Plans Coal Storage for Employees

The Connecticut Co., of New Haven, has started a plan for the distribution of domestic coal to its employees which undoubtedly will bring about the getting of coal for the winter of 1923-24 into the individual bins of the employees during the summer months. They have sent out a questionnaire to each employee asking for the kind, size, quantity, name of dealer who has been supplying them coal in the past, so that the same dealer may be employed if desired, place of delivery and date of taking the full supply.

After replies have been received and arrangements made for the delivery, an agreement will be made between the company and the individual, authorizing the deductions of the proportion of the exact coal bill from the payroll.

New River To Have 90,000-Hp. Steam-Generating Plant

In order to increase its traffic capacity and to secure important operating economies, the Virginia Railway has decided to electrify 213 miles of its track lying between Roanoke, Va., and Mullens, W. Va. This undertaking will involve the expenditure of $15,000,000. The order for the electric locomotives, power house, transformer stations, and other apparatus has been awarded to the Westinghouse Electric & Manufacturing Co., and forms the largest railroad electrification contract ever placed. The division to be electrified crosses the Allegheny Mountains. The alternating current single-phase system will be used.

The largest steam locomotives now in use on the Virginian are of the articulated Mallet type, with twenty driving wheels and four cylinders. Three of these huge engines are used to move 5,500-ton trains over the grades, but their combined power does not exceed 7,000 hp., and their speed on the grades is only 7 miles an hour. The new electric locomotives, however, developing 20,000 hp. per train, will haul 9,000-ton trains up the grades at the rate of 14 miles an hour, and it will be entirely practical in the future to further increase this power so that 12,000-ton trains can be handled at the same speed.

A feature of the electric locomotives will be the use of regenerative braking on the down grades. This method of braking will not only reduce the wear on the brake shoes and wheels and improve operation, but will also save 15,000,000 kw.-hr. of energy per year.

Power for operation will be supplied by a 90,000-hp. steam-generating plant to be erected on the New River. This will supply 88,000-volt current to the main transmission line. For use on the trolley wire from which the locomotives will draw their power, this high-voltage current is to be stepped down to 11,000 volts by transformer stations placed at regular intervals along the line. On the locomotives this is reduced to a low value for the operation of the motors.

World Power Conference Program

L. B. Atkinson, executive head of the Cable Makers' Association, Roger T. Smith, electrical advisor to the Great Western Railway, and Sir John Snell, one of the governmental Electrical Commissioners, have been nominated by the Institution of Electrical Engineers to assist in the preparation of the final program for the World Power Conference at London in 1924.

Boulder Lake Power Plant Started

Preliminary work has begun on the site for the superpower plant that the Doherty interests will build at Boulder Lake, four miles east of Boulder, Colo., to supply the requirements of the Western Light & Power Co. and the Denver Gas & Electric Light Co. The first unit of the plant will have a generating capacity of 20,000 kw. and will represent an investment of $4,000,000. This is the first of an estimated total investment of $12,000,000.

A tentative sketch of the first unit of the plant shows a building 330 by 140 ft. A feature of this plan is a smokestack 350 ft. high, with an inside measurement at the top of 14 ft. The height of the stack, it is explained, bears a direct relation to the reduced air pressure at this altitude and the necessity of maintaining intensive draft under the boilers. If this tentative program is carried out, the stack at the Boulder Lake plant will be one of the tallest structures in the West.

When the present lake has been enlarged, the plant will have a picturesque setting. Viewed from the Denver Boulder highway, it will appear to rise out of an expanse of water covering 400 acres, and Long's Peak, although fifty miles away will mirror itself in the placid waters of Boulder Lake.

Coal Production in the Ruhr

In the Ruhr there is now raging the most bitter economic war known to history. There is being fought the belated but final battle of the war, and Germany is putting all her strength and a large part of her remaining substance into it.

France owned 35 per cent of the iron deposits of Europe before the war. Germany 25½ per cent. By territorial changes incident to peace France's share rose to 53 per cent and Germany's dropped to 7½ per cent. Germany is still owner of 33½ per cent of Europe's coal resources, while France's share is 2.3 per cent and with the Sarre basin 4.4 per cent. Germany has in this respect still the leading position in Europe, further strengthened by her large brown-coal deposits. The temporary possession of the Sarre district secures for France an additional quantity of eleven million tons annually, and the German coal tribute is fifteen million tons. Both will, however, come to an end some day.

In 1918, 55 per cent of the iron ore consumed in the Ruhr came from abroad. Germany can still procure supplies from this source, but she is not dependent on them. On the other hand, the dependency of French iron on German coal is obvious. France has no other alternative than either to reduce the output of iron ore or to ship the surplus to the German blast furnaces.

The Ruhr district has the most dense and most elaborate railroad system in Europe. All junctions in the heart of the district are controlled electrically from one central station. Modern automatic signals and controlling devices are employed throughout which, when in working order, constitute great facilities. Out of gear they become dead weight. The crudest makeshift methods were resorted to, and this condition prevailed also in regard to the hoisting and conveying machinery. Essential parts were removed when seizure threatened and hidden in the pits. Repair was out of the question without close acquaintance with the design and without capacious repair shops. So far, the measures of resistance taken have been very effective. The occupying force has restricted its energy to putting one railroad line across the northern part of the district in commission. Loading operations are still performed in the most primitive fashion by shovels, baskets and wheelbarrows. Only a fraction of the quantity of coal shipped to France prior to the occupation can be removed now from the vast stocks of coal and coke accumulated at pitheads and nearby spaces. The German government has evidently secured the first trick and is still holding some master cards in its hands.

New Equipment Being Installed

Work has begun on the installation of a 30,000-kw. turbine at the Acme Plant of the Toledo Edison Co. The Acme plant was completely described in *Power*, Sept. 20, 1921. There is also being installed a 20,000-kw. turbine at Lorain, Ohio, and a 10,000-kw. turbine for the Empire District Electric Co., at Joplin, Mo. To provide for more efficient generation and also to take care of an increasing load, a new plant is being built at Sedalia, Mo., and additional generating facilities are being provided at Salina, Kan., and at Amarillo, Texas.

Columbia River Investigation To Start Soon

Definite steps to begin the investigation authorized by Congress of the Columbia River Basin in the State of Washington with respect to the reclamation of several million acres of lands included in it, will be taken by the Interior Department next week.

Assistant Secretary of the Interior F. M. Goodwin, a member of the commission named by Secretary of the Interior Work to conduct the investigation, will leave for Spokane, Wash., within the next few days where he will meet Director A. P. Davis of the Reclamation Service, another member of the Commission, for the purpose of outlining plans for the field work. Mr. Goodwin will make a short trip over some of the routes of the Basin, where it is proposed to construct diversion dams and other works, after which he will go to the Umatilla Rapids district in the State of Oregon, Congress having appropriated the sum of $50,000 for the purpose of investigating its reclamation possibilities.

Production of British and American Coal

The elimination of the Ruhr coal from European markets is causing an added demand of more than 2,000,000 tons of British coal per month for Continental consumption and in consequence the price of coal is rising and while British loss of production due to Easter holidays is significant, the gain in production is still steadily increasing. For the week ended April 14, 5,776,000 tons were mined, the largest weekly output in the last four years.

During February France imported 1,375,000 tons of British coal, Italy 675,000, Germany, 1,000,000, and Belgium 425,000 tons. In March France and Germany each imported 1,800,000 tons, or a combined excess of 1,750,000 tons above the 1922 monthly average, according to government reports.

While the situation in the Sarre basin is somewhat improved by the return of some of the strikers to work, production is still below normal. It is apparent that as long as the Ruhr deadlock continues, Western Europe is facing a probable net shortage of coal, which provides an opportunity for shipments from the United States.

In America the total output of coal for the week ended April 21, was estimated at 2,065,000 net tons of anthracite and 10,244,000 of soft coal, there being a decrease in the weekly output from the preceding week in soft coal of 157,000 tons.

Nationalize Coal Mines, Demands British Labor

Nationalization of the coal industry of Great Britain is the object of legislation backed by the Labor Party, which has been recently introduced in the British Parliament. The proposed law, known as the "Nationalization of Mines and Minerals Bill," provides for: Establishment of a mining council consisting of a minister of mines and 20 members; for ten mine-purchase commissioners, who will assess mining properties, prices so fixed to be paid in mines purchase stock and for this purpose the Treasury shall direct the creation of a new capital stock to be called guaranteed states mines stock; control over every department of the mining industry, including the marketing and sale of coal; appoint mining councils and pit councils; have power to appoint a fuel consumers' council.

Boiler and Machine Shops Show Gain in Products

The Department of Commerce announces that reports made by boiler shops, foundries and machine shops show products manufactured to the value of $1,565,527,000 in 1921, as compared with $2,289,251,000 in 1919 and $866,545,000 in 1914—a decrease of 32 per cent from 1919 to 1921, but an increase of 81 per cent for the seven-year period, 1914 to 1921.

Quebec Lets Two Contracts for Water Power

Contracts for two important water-power projects have recently been let in Quebec. One of these is in connection with the rapids of the Grande Decharge of Lake St. John, while the other is a 15,000-hp. development on the Outarde River on the north shore of the coast.

It is expected that within five years a minimum of 230,000 hp. will be developed at a cost of $12,000,000. There are said to be possibilities for developing 1,000,000 hp. from the stretch of rapids extending from Lake St. John to the head of navigation of the Saguenay, and in time this figure will undoubtedly be reached. This will place the Province of Quebec at the head of all provinces producing electric power in Canada, says Consul E. H. Dennison in a report to the United States Department of Commerce.

Boiler Code Committee Honor Memory of Joseph H. McNeill

The Boiler Code Committee of the A.S.M.E., profoundly impressed by the news of the sudden taking away of their friend and associate, Joseph H. McNeill, and desirous of recording their esteem for him and of expressing the sympathy that they feel for his family, unanimously adopted at their meeting on April 27, the following:

Resolved, That in the death of Joseph H. McNeill, the application of engineering knowledge and experience to the design, construction and operation of steam boilers has lost one of its foremost American advocates and administrators.

That, when this Committee took up the task assigned to it, we found in Mr. McNeill a pioneer worker in the field; and in the service which he had already accomplished for the Commonwealth of Massachusetts a precedent which has been of great value in our own work;

That in our contact with him we have found him a competent engineer, a conscientious official, a staunch and true friend and a genial, lovable companion;

That, we sympathize with those still more closely bound to him by ties of kinship and affection, and direct that a copy of this resolution be conveyed to his family and spread in full upon the records.

Farewell Dinner Given To John Lane

On the evening of April 26 a farewell dinner was tendered John W. Lane by the combined N. A. S. E., associations of Chicago, the occasion being his voluntary retirement from the editorship of the *National Engineer*. Upward of 200, including a delegation from Wisconsin, did honor to the departing guest. Community singing and the famous quartette of No. 1 enlivened the occasion. Alfred Johnson served as toastmaster. Regrets and best wishes were expressed by a number of speakers prominent in association work, and as a finale Mr. Lane was presented with a handsome suitcase, for use on an extended trip to Europe.

After twenty-one years of continuous service as editor of the paper, Mr. Lane is to take a well-earned rest. It is his intention to leave on May 5 by way of Montreal, for London and to spend several months in Great Britain and perhaps on the Continent. Fred Raven, national secretary of the association, on a two months' leave of absence, is to accompany him. At the dinner Fred also received a suitcase. His friends desired to show the proper spirit, but requested him to use the grip in bringing back their spirits to them from the lands of his travel.

Thomas Thurston, who has been assistant editor of the *National Engineer* for a number of years, will succeed Mr. Lane.

Only 25 per cent of the members of the American Society of Mechanical Engineers in the State of New York, as indicated by a count of the questionnaire sent out by the Society, have applied for a license to practice professional enginering.

Society Affairs

The **Hartford Branch A.S.M.E.** will hold a meeting at the City Club on May 14. Representatives of the Metal & Thermit Co. and the Lincoln Electric Co. will speak on "Welding."

The **Chicago Section** of the **A.S.M.E.** will hold their annual meeting with election of officers on May 15. It will be ladies' night and Dean M. E. Cooley, of the Michigan University, will speak on "The Engineer as a Citizen."

A **Boston Chapter** of the **Society of Industrial Engineers** was inaugurated on Wednesday, April 25. Officers were elected as follows: President, Matthew Porosky; secretary, H. C. Barker; treasurer, C. O. Wellington; members of Executive Council, R. G. Wells and W. E. Kemp.

The American Society of Mechanical Engineers has arranged for an excursion en route to Montreal for members from the West, to visit the Queenstown-Chippawa Power Plant and the substation of the Hydro-Electric Power Commission of Toronto. A visit to the power developments and paper mills on the St. Maurice River at Grand Mere and Shawinigan Falls, following the meeting, has also been planned.

The Society of Heating and Ventilating Engineers will have the following program at their meeting at the Drake Hotel, Chicago, Ill., May 21-23: Discussion on Code for Heating and Ventilating Equipment; Report of Committee on Research; address by Director F. Paul Anderson; Paper, "Ozone," by W. P. Yant, G. W. Jones and F. C. Houghten; "Dust Research," by S. H. Katz, "Further Observations in Dust Determinations," by Margaret Ingels; "Heat Transference and Combustion in a Small Domestic Boiler," by John Blizard, W. C. Myler, Jr., J. K. Seabright and C. P. Yagloglou; "Study of Physiological Reactions in Still Air," by W. J. McConnell, F. C. Houghten and C. B. Yagloglou; "Determination of the Comfort Zone," by F. C. Houghten and C. P. Yagloglou; "Insulation of Cold Surfaces to Prevent Sweating," by L. L. Barrett; "Hot-Water Heating at Chase Metal Plant, Waterbury, Conn.," by J. E. Williams; "Temperatures at Registers of Warm Air Heating Systems," by V. R. Day; "District Heating Progress in Boston," by D. S. Boyden; Discussion of Report of New York State Commission on Ventilation, Report on Ventilation Code of Wisconsin, "Heating and Ventilation in Chicago Public Schools."

Personal Mention

Walter Tallmadge has associated himself with the Donnelly Systems Co., 9 Murray St., New York City.

F. W. Dean, of Wheelock, Dean & Bogue, Inc., Boston, has just returned from a sojourn in Bermuda.

G. H. Froebel, formerly manager of the Houston, Texas, branch office, has been appointed manager of the Marine Sales Department of the Westinghouse Electric & Manufacturing Co.

James E. Lea, of the Lea Recorder Co., Ltd., Manchester, England, and inventor of the Lea V-notch meter for water measurement, is in America seeking to introduce the Lea coal meter.

Joseph Pope is now doing steam-power station research work for Stone & Webster, Inc., of Boston. His former position as head of the Power Station Betterment Division has been filled by Walter E. Balcke.

Claude Allen, for several years New York representative of the Garlock Packing Co., has been put in charge of the company's business on the west coast, with headquarters at San Francisco. Before coming to New York he was in charge of the Boston office.

E. B. Powell, consulting engineer of Stone & Webster, Inc., returned from France on the "Paris" on April 21. Owing to a severe illness he was confined to the American Hospital at Neuilly, France, for three months, but is now much improved in health.

Charles H. Bigelow, who for the last ten years has acted as chief mechanical engineer for the Millville (N. J.) Manufacturing Co., designing and constructing several industrial buildings and a modern power plant, has accepted the position of plant engineer for the Spicer Manufacturing Corp., South Plainfield, N. J., manufacturer of universal joints and clutches for automobiles.

Business Notes

The Louis Allis Co. announces that the Commercial Electric Sales Co., 1322 Empire Bldg., Birmingham, Ala. is to be its Alabama agent.

The Superheater & Engineering Co., moved its offices from 1816 Woolworth Building to 103 Park Ave., New York City, on May 1.

S. F. Bowser & Co., Inc., Fort Wayne, Ind., have announced the appointment of Paul W. Lawther as sales manager for Texas with offices in Dallas, Texas.

S. P. Wright & Co., Butte, Mont. district representative of the Conveyors Corporation of America, 326 West Madison St., Chicago, have moved their offices from 109 East Broadway to 812 East Iron St., Butte, Mont.

Brady Conveyors Corp., of Chicago, some time ago purchased the manufacturing rights of the Brady steam jet ash conveyor, flat arch, ash storage hopper and the Cox-Fulton and Simplex stokers. This company is functioning as usual and is not affected in any way by the recent retirement of the James A. Brady Foundry Co.

The **Liptak Fire-Brick Arch Co.,** of Chicago, manufacturer of Liptak double suspension arches and the Liptak interlocking fire brick wall, has recently secured the services of C. R. Matheny, who will act as district manager, with offices at 50 Church St., New York. He was formerly connected with the Republic Flow Meters Co.

The United Machine & Manufacturing Co., 1208 Camden Ave., S. W., Canton, Ohio, manufacturer of the Harrington Traveling Grate Stoker, the Industrial Underfeed Stoker and the Mosher Separator, is taking over the Canton-Detroit Foundry Co., which is to be utilized as a foundry department under the present management.

The Canadian Ingersoll-Rand Co., Limited, 260 St. James St., Montreal, are placing on the Canadian market a complete line of Straight Line and Duplex Ammonia Compressors for steam, belt, electric-motor or oil-engine drive. These machines correspond with those manufactured in this country by the Ingersoll-Rand Co., New York.

Donnelly Systems Co., 9 Murray St., New York City, has recently placed Webster Tallmadge in charge of the sales and service departments. The sales department will be extended and the service department developed. Mr. Tallmadge was for a number of years connected with the Westinghouse company and recently held the position of technical manager with the Vacuum Oil Co. The Donnelly Systems Co. plans to extend its activities, particularly along the line of temperature control for industrial plants, for which a new method of control has been developed.

Coming Meetings

American Boiler Manufacturers' Association; H. N. Covell, 191 Dikeman St., Brooklyn, N. Y. Thirty-fifth Annual Convention at Hot Springs, Va., June 4-6.

American Order of Steam Engineers; W. J. Brice, 2126 S. 57th St., Philadelphia, Pa. 37th Annual Convention at Reading, Pa. (instead of Philadelphia, as formerly announced), June 4-8.

American Society for Testing Materials; C. L. Warwick, 1315 Spruce St., Philadelphia, Penn. Annual meeting at Atlantic City, N. J., June 25-30.

American Society of Heating & Ventilating Engineers; O. W. Obert, 29 West 39th St., New York City. Semi-annual meeting, Chicago, May 21-23.

American Society of Mechanical Engineers, 29 W. 39th St., New York City. Spring meeting at Montreal, Canada, May 28-31.

American Water Works Association, 163 W. 71st St., New York City. Annual meeting at Detroit, Mich., May 21-25.

Canadian Association of Stationary Engineers; Gordon C. Keith, 51, Wellington St. West, Toronto, Canada. 34th Annual Convention at Toronto, Canada, June 25-28.

Electric Power Club; S. N. Clarkson, Kirby Bldg., Cleveland, Ohio. Annual meeting at Hot Springs, Va., June 11-14.

International Railway Fuel Association; J. G. Crawford, 702 East 51st St., Chicago. Annual meeting at Cleveland, Ohio, May 21-24.

National Association of Manufacturers; George S. Boudinot, 50 Church St., New York City. Annual conference, New York City, May 14-16

National Association of Stationary Engineers; Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual convention and exhibition at Buffalo, N. Y., Sept. 10-15. Annual conventions and exhibitions of the State Associations scheduled as follows: **Indiana,** at La Fayette, Ind., May 28-29; T. J. Kelly, 2704 South St., La Fayette, Ind. **New Jersey,** at Trenton, N. J., June 1-3; Samuel Clark, 67 Columbia St., Jersey City, N. J. **Illinois,** at Springfield, Ill., June 6-9; F. W. Raven, 417 South Dearborn St., Chicago. **New England States Association,** at Manchester, N. H., July 12-14; James F. Morgan, 53 Devonshire St., Boston, Mass. **Iowa,** at Mason City, Iowa, June 11-14; Abner Davis, Room 16, Waterhouse Block, Cedar Rapids, Iowa. **Ohio,** at Dayton, June 21-23; J. H. Weaver, 22 N. Gift St., Columbus, Ohio. **Pennsylvania,** at Buffalo, Sept. 9-10; J. N. Calvert, Crafton Sta., Pittsburgh, Pa. **New York,** at Buffalo, Sept. 9-10; W. T. Meinzer, Third St., near Warburton, Bayside, L. I., N. Y. **Minnesota,** at Duluth, Aug. 8-10; C. A. Nelson, 800 22d Ave. N. E., Minneapolis. **Connecticut,** at Waterbury, June 29-30; L. Van Der Eyk, 251 Wood St., Waterbury.

National District Heating Association, D. L. Gaskill, Greenville, Ohio. Annual convention at Cedar Point, Ohio, June 20-23.

National Electric Light Association; M. H. Aylesworth, 29 West 39th St., New York City. Annual meeting at New York City, June 4-8.

Smoke Prevention Association; Frank Chambers, 111 N. Dearborn St., Chicago, Ill. Convention at Minneapolis, Minn., June 19-22.

Society for the Promotion of Engineering Education; F. L. Bishop, University of Pittsburgh, Pittsburgh, Pa. Annual meeting at Cornell University, Ithaca, N. Y., June 20-23.

Stoker Manufacturers' Association; J. G. Worker, Eau Claire, Wis. Annual convention at Lake Placid Club, Essex Co., N. Y., May 29-31.

Universal Craftsman Council of Engineers; Jerome Marshall, 404 West Elm St., Rochester, N. Y. Convention at Rochester, N. Y., Aug. 7-11.

Trade Catalogs

Crane, 10-ton Crawler No. 2 — McMyler-Interstate Co., Bedford, Ohio. Bulletin No. 46 presents specifications, photos and drawings of their new No. 2 steam-operated convertible crane.

Turbines—Kerr Turbine Co., Wellsville, N. Y. Bulletin No. 103 is a pamphlet containing descriptions and photographs of the place where the turbines are made and the different processes involved in the making of them.

Enco Oil Burning Instruction Book—Engineer Co., 17 Battery Place, New York City. This small instruction book contains important information as to the general requirements for efficient operation, management, casualties, safety precautions, handling, installations and equipment and properties of fuel oil for use under steam boilers. A good index, illustrations and charts, as well as the text, make this an invaluable book for the operating engineer of oil-fuel installations.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Apr. 23, 1923	Apr. 30, 1923
Pool 1,	New York	$3.50–4.25	$3.75–4.25
Smokeless,	Columbus	4.00–4.50	3.75–4.50
Clearfield,	Boston	2.25–3.25	2.00–3.25
Somerset,	Boston	2.85–3.50	2.50–3.50
Kanawha,	Columbus	2.00–2.50	2.00–2.50
Hocking,	Columbus	1.85–2.20	1.85–2.20
Pittsburgh No. 8	Cleveland	2.10–2.25	2.10–2.25
Franklin, Ill.,	Chicago	3.00–3.25	3.00–3.25
Central, Ill.,	Chicago	2.00–2.25	2.00–2.25
Ind. 4th Vein,	Chicago	2.75–3.00	2.75–3.00
West Ky.,	Louisville	2.00–2.25	1.85–2.10
S. E. Ky.,	Louisville	2.50–2.75	2.50–3.00
Big Seam,	Birmingham	2.00–2.25	2.00–2.25

FUEL OIL

(f.o.b. city unless otherwise specified)

New York—May 3, Port Arthur light oil, 22@25 deg. Baumé, 5⅜c. per gal.; 30@35 deg., 5¾c. per gal., f.o.b. Bayonne, N. J.

Chicago — April 28, 24@26 deg. Baumé, $2.12 per bbl.; 32@36 deg., $2.33 per bbl. tank cars.

Philadelphia—April 23, 26@28 deg. Baumé, Oklahoma, $1.10@$1.20 per bbl.; 30@34 deg., Oklahoma (group 3), 3⅛@3⅜c. per gal.; 16@20 deg. Seaboard, $1.75@$1.90 per bbl.

St. Louis—April 24, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.90 per bbl.; 26@28 deg., $1.95 per bbl.; 28@30 deg., $2 per bbl.; 30@36 deg. gas oil, $2.20 per bbl.; 36@40 deg., distillate $2.35 per bbl.

Pittsburgh—April 28, f.o.b. local refinery, 30@34 deg., fuel oil, 5¾c. per gal.; 36@40 deg., fuel oil, 6c. per gal.; 34 deg., neutral, 9½c. per gal.

Dallas—April 28, f.o.b. local refinery, 26@30 deg., $1.59 per bbl.

New Plant Construction

PROPOSED WORK

Calif., Long Beach—The Concord Mansions Co. is having plans prepared for the construction of a 10 story, 50 x 150 ft. apartment building on Atlantic St. Estimated cost $600,000. E. Bergstrom, City Natl. Bank Bldg., Los Angeles, Archt. Equipment detail not reported.

Calif., Los Angeles—A. C. Blumenthal, Loew State Bldg., is having plans prepared for the construction of a 12 story, 120 x 293 ft. store and office building on Broadway. Estimated cost $4,000,000. S. T. Norton, Pacific Finance Bldg., Los Angeles, Archt. Equipment detail not reported.

Calif., Los Angeles—F. Harper, 672 Benton Way, plans to build a 12 story apartment hotel on a 123 x 150 ft. site on Benton Way near 7th St. Cost will exceed $650,000.

Calif., Los Angeles—D. Hart, c/o O. H. Neher, Archt., Marsh Strong Bldg., is having plans prepared for the construction of a 12 story theatre and office building on Olive St. near 7th St. Cost will exceed $500,000. Equipment detail not reported.

Calif., Los Angeles—The West Coast Theatre Co., c/o Milwaukee Bldg. Co., Archts., Wright-Callendar Bldg., is having preliminary plans prepared for the construction of a store and theatre building near Washington and Vermont Sts. Cost will exceed $500,000.

Calif., Oakland—The Providence Hospital Assn., c/o R. A. Herold, Forum Bldg., Sacramento, Archt., is having sketches completed for the construction of a 4 story hospital on Webster and Summit Sts. Estimated cost $500,000. Noted July 18, 1922.

Calif., Paradise—Paradise Irrigation Dist. is having plans prepared for a battery of hydraulic rams, probably 24 in., to cost about $5,000 each, a small centrifugal pump will be required. S. J. Norris, Oroville, Engr.

Calif., San Francisco—A. Pantages, Pantages Theatre Bldg., plans to build a theatre. Cost will exceed $500,000.

Calif., San Jose—E. Tropp and W. C. Crittenden, 129 Sutter St., San Francisco, plan to build a 16 story hotel to contain 500 rooms, here. Engineer or architect not selected.

Calif., Willows—The San Jacinto Irrigation District, here, is having plans prepared for a drainage system involving construction of pumping plants, 15 wells and deep well turbine pumps, 18 and 20 in., with power lines, motors, etc. J. F. Ryan, Consult. Engr.

Conn., Hartford—H. S. Hooker, c/o Shape, Bready & Peterkin, Archts., 50 East 42nd St., New York, is having preliminary plans prepared for the construction of a 4 story, 200 x 250 ft. apartment building, including steam heating system, on Asylum and Gillet Sts., here. Estimated cost $500,000.

Conn., New Haven—The Connecticut National Guard, c/o Quartermaster General, State Armory, Broad St., Hartford, plans to build an armory. Estimated cost $500,000. Engineer or architect not selected.

D. C., Wash.—The Washington Opera Co., is having plans prepared for the construction of an opera house, bank and office building on Dupont Circle. Estimated cost $1,000,000. G. O. Totten, Jr., 800 17th St. N. W., Archt. Equipment detail not reported

Fla., Orlando—The Orlando Mfg. Co., P. O. Box 1256, E. J. Jenkins, Pres., is in the market for complete machine shop equipment and motors from ¼ to 7 hp.

Fla., Tampa—The Consumers Ice Co., Polk and Marion Sts., is having plans prepared for the construction of a 30 ton ice plant at Sulphur Springs. Estimated cost $65,000. Engineer or architect not announced.

Ind., Monticello—The city will receive bids until May 15, for two 300 g.p.m. motor driven centrifugal pumps and one 800 g.p.m. motor driven fire pump for city waterworks.

Kan., Horton—The city, I. Griffith, City Clk., will receive bids until May 29 for the construction of an earth dam with spillway and intake, flow line, clear water storage reservoir and a high service pump station. Black & Veatch, Mutual Bldg., Kansas City, Mo., Engrs.

Kan., Smith Center—S. C. Walker is receiving bids for the construction of a 3 story, 48 x 100 ft. warehouse and cold storage plant. Estimated cost $40,000. W. E. Glover, Stormont Bldg., Topeka, Archt.

Kan., Smith Center—The city, J. M. Perry City Clk., will receive bids until May 16 for furnishing pumping equipment for the low service pumping station at Gaylord and the filter plant, and high service station here. (1) Two deep well type triplex pumps, capacity 180 g.p.m., each against a head of 450 ft., complete with motors, starters, chain drives, etc., one automobile type gasoline engine with all accessories for driving either of above pumps. (2) Two horizontal split case, motor driven centrifugal pumps, 180 g.p.m., each against a maximum head of 175 ft. complete with motors, starters and accessories. (3) One horizontal split case gasoline engine driven centrifugal pump, capacity 1,350 g.p.m., against a head of 40 ft. complete with automobile type gasoline engine, chain drive and accessories. (4) One horizontal split case gasoline engine driven centrifugal pump, capacity 500 g.p.m., against a total head of 250 ft. with all accessories. Black & Veatch, Mutual Bldg., Kansas City, Mo., Engrs.

Kan., Topeka—The Topeka Hotel Co. is receiving bids on the general contract for the construction of a 10 story, 100 x 150 ft. hotel on 7th and Jackson St. Estimated cost $600,000. T. W. Williamson & Co., 312 Central Natl. Bank Bldg., Topeka, Archts. Noted Mar. 20.

Ky., Louisville—U. S. Engineers Office, L. S. Ehrman, Purch. Agt., is in the market for one 75 hp. 550 volt 450 r.p.m. vertical shaft motor with control panel.

Md., Baltimore—S. T. Williams, 223 North Calvert St., is in the market for a 200 or 250 hp. Corliss or similar type engine with boilers, preferably water tube boilers (new or used).

Mass., Rutland—Office of Quartermaster Genl., 2306 Munitions Bldg., Wash., D. C., will receive bids May 31 for the construction of ten buildings, including central heating plant with distribution main and extensions to water, sewer and light lines, here.

Mich., Birmingham—The village, H. E. Clark, Clk., is having plans prepared by the village engineer for the construction of a well, 18 in. in diameter or larger, also pumping equipment, consisting of motor driven centrifugal or other type pump, capacity 400 g.p.m. Estimated cost $15,000.

Mich., Detroit—Bonnah & Chaffee, Archts., 528 Farwell Bldg., are receiving bids and will open same about May 15 for the construction of 1 and 2 story, 80 x 150 ft., 80 x 250 ft. and 50 x 150 ft. steel mill buildings, including boilers and equipment for heating plant, on Fort Blvd. for the Detroit Steel Co., 6189 Hamilton Ave., W. C. Schrage, Pres. The contract for motors has been awarded to the General Electric Co., River Rd., Schenectady, N. Y.

Mich., Detroit—The Dept. of Public Wks., will receive bids until May 14 for the construction of a 53 x 131 ft. pumping station, including 60 x 120 ft. pump pit, 40 ft. deep, also suction well. J. W. Reid, City Hall, City engr.

Mo., Jasper—A. C. Moore, Engr., 222 Independent Bldg., Joplin, will soon receive bids for the construction of a waterworks system, estimated cost $40,000, also an electric light plant, estimated cost $5,000. Bonds have been sold for the projects.

Mo., Kansas City—J. Fairweather, 1005 Scarritt Bldg., representing owner, plans the construction of a 3 story, 100 x 125 ft. hospital. Estimated cost $400,000. Architect not selected.

Mo., Oran—The city plans election May 15, to vote $65,000 bonds for municipal electric light plant and waterworks improvements. Electric light work to consist of one 50 kva. 75 hp. oil engine, air compressor, 10,000 gal. oil storage tank, 50 kva. generator belted type, switchboard panel, 45 street lights, 190 30 ft. poles, etc., estimated cost $23,000. Waterworks to consist of mains, hydrants, valves, 60,000 gal. reservoir or 60,000 steel tank or tower, deep well pump and motor, etc., estimated cost $42,000. W. A. Fuller, 1917 Railway Exch. Bldg., St. Louis, Engr. Noted April 10.

Mo., Piedmont—The city is having plans prepared for the construction of a waterworks and distribution system, also pumping units for 30,000 gal. W. A. Fuller, 1917 Railway Exch. Bldg., St. Louis, Engr.

Mo., St. Louis—The Milly Realty Co., Princess Hotel, plans the construction of a 12 story, 73 x 90 ft. hotel on Market St. Estimated cost $750,000. Architect not selected.

Mo., St. Louis—The St. Louis Maternity Hospital, 4518 Washington Ave., is raising funds and plans to build an 8 story hospital to contain 120 beds, on Kingshighway. Estimated cost $500,000.

N. J., Princeton—The Princeton Inn Co., c/o A. J. Thomas, Archt. and Engr., 15 East 47th St., New York, will soon receive bids for the construction of a hotel on Alexander St. Estimated cost $350,000. Equipment detail not reported. Noted Dec. 7.

N. Y., Charlotte—The Fruit Growers Syndicate, A. E. Kelley, Pres., is having plans prepared for the construction of a 3 story, 120 x 140 ft. cold storage warehouse on Fleming Rd. Cost between $150,000 and $160,000. J. Strobel, 52 Lake View Park, Rochester, Archt.

N. Y., Medina—The Western New York Utilities, Inc., is in the market for a generator and auxiliary equipment for power plant at Waterport.

N. Y., New Rochelle—M. Geviner, c/o H. R. Burroughs, Engr. and Archt., 70 East 45th St., New York, is having plans prepared for the construction of a theater and apartment building on Division St. here. Estimated cost $500,000. Equipment detail not reported.

N. Y., New York—F. J. Willetts, c/o G. F. Pelham, Engr. and Archt., 200 West 72nd St., will build a 9 story apartment building at 142 East 49th St. by separate contracts under architect's supervision. Equipment detail not reported.

N. Y., Niagara Falls—The Niagara Falls Arena Co., Inc. plans the construction of an ice rink, including 60 ton ice plant on Victoria Ave. Estimated cost $175,000.

N. Y., Rome—The Manufacturers Power Corp., c/o M. J. Larkin, 106 East Dominick St., plans to build a power house on the Barge Canal, and transmission lines. Estimated cost $100,000.

N. Y., Wilson—The Niagara County Preserving Co. is in the market for one 125 hp. A. C. 220 volt, 3 phase, 25 cycle, 750 r.p.m. slip ring type electric motor.

N. C., Fayetteville—The Chamber of Commerce is promoting a hotel company, not yet organized, and is having plans prepared for the construction of an 8 story hotel. Estimated cost $250,000. W. G. Rogers, Trust Bldg., Charlotte, Archt. Equipment detail not reported.

N. C., Lenoir—B. V. Booth, M. R. Warren and C. Guenter, Danville, Va., plan the development of kaolin and aluminum mines, including a small tunnel, also development of water power, 8 mi. from here. Estimated cost $50,000. C. Guenter, Danville, Va., Engr.

N. C., Lexington—C. M. Wall & Son, C. M. Wall, Mgr., is in the market for complete machinery and equipment for model box plant, including 250 hp. power plant.

N. D., Fargo—The Union Light, Heat & Power Co. plans the construction of a double water main from Red River of the North above dam to plant and the installation of a turbine engine unit, condenser type. Estimated cost $300,000.

Ohio, Cleveland—The Bd. Educ., F. G. Hogen, Dir., is having plans prepared for the construction of a 6 story high school to contain 60 class rooms, including steam heating plant, on University Circle. Estimated cost $3,000,000. W. R. McCormack, East 65th St. and Rockwell Ave., Cleveland, Archt.

Ohio, Cleveland—The Bd. Educ., c/o F. G. Hogen, Dir., East 6th St. and Rockwell Ave., is having plans prepared for the construction of a 4 or 6 story Bd. of Educ. office building, including steam heating system, on East 6th St. and St. Clair Ave. Estimated cost $500,000. W. R. McCormack, c/o owner, Archt.

Ohio, Cleveland—The Broadhead-Garrett Co., 2835 East 51st St., J. C. Rash, Asst. Secy., will receive bids until about May 15 for the construction of a 300 x 360 ft. warehouse, 40 x 40 ft. dry kiln, 40 x 50 ft. planing mill, boiler house, office and garage building, on East 71st St. along tracks of Belt Line R.R. Estimated cost $200,000. Best & Hoefler, 6523 Euclid Ave., Cleveland, Archts. Noted Apr. 10.

Ohio, Cleveland—The Cleveland Homeopathic Medical Society, c/o L. E. Siemon, Pres., 423 Erie Bldg., plans to build a group of hospital buildings, including steam heating system at 724 Huron Rd. Estimated cost $2,000,000. Engineer or architect not selected.

Ohio, Cleveland—The Cleveland Public Library Bd., J. White, Kinney Levan Bldg., Pres., will receive bids until May 28 for the construction of a 4 story, 199 x 216 ft. library building, including steam heating system, on East 3rd St. and Superior Ave. Estimated cost $4,000,000. Walker & Weeks, 1900 Euclid Ave., Cleveland, Archts. Noted Oct. 31.

Ohio, Cleveland—The Euclid Avenue Baptist Church, c/o W. W. Bustard, pastor, 13679 Fairmount Blvd., is having plans prepared for the construction of a 9 story, 134 x 360 ft. church and office building on East 18th St. and Euclid Ave. Estimated cost $1,000,000. Walker & Weeks, 1900 Euclid Ave., archts. Equipment detail not reported.

Ohio, Columbus—The Continental Clay Co. Atlas Bldg., W. B. Ferris, Pres., is in the market for a 300 hp. steam engine and boiler; for plant at Zanesville.

Ohio, Portsmouth—The Hurth Hotel Co. is having plans prepared for the construction of a 6 story, 90 x 150 ft. hotel on 3rd and Chillicothe Sts. Estimated cost $350,000. C. C. Taylor, Portsmouth, Archt. Equipment detail not reported.

Ohio, Springfield—The Masonic Temple Assn. is having plans prepared for the construction of a 5 story, 100 x 150 ft. masonic temple, including auditorium to seat 4,500. Estimated cost $600,000. Miller & Reeves, 20 South 3rd St., Columbus, Archts. associated with H. D. Smith, Archt., 50 East Town St., Columbus. Equipment detail not reported.

Okla., Maysville—The town voted $35,000 bonds and will probably receive bids about June 1 for the construction of a waterworks system, tank, tower and pump. The Mackintosh Co., 722 1st Natl. Bank Bldg., Oklahoma City, Engrs.

Okla., Sand Springs—The Consumers Ice Co. is having plans prepared for the construction of an ice plant. Estimated cost $75,000. Private plans.

Okla., Tonkawa—The city, E. E. Davis, City Clk., will receive bids until May 15 for water works improvements, consisting of furnishing and erecting one 500 g.p.m. triplex plunger pump, driven by 60 hp. semi-diesel engine; construction of engine pit and placing of concrete floors in present well; furnishing and laying about 289 ton of 8 in. Class B and 120 ton of 10 in. Class B c.i. pipe, also 5 ton of 6 in. pipe; special castings and valves. Ruckel Eng. Co., Hutchinson, Kan., Engrs.

Pa., Phila.—The University Club, 1510 Walnut St., is having plans prepared and will soon receive bids for the construction of a 23 story, 40 x 100 ft. club building, including steam heating system. Estimated cost $1,000,000. Simon & Simon, 249 South Jumper St., Archts.

Pa., Titusville—A. J. Benton, 331 East Walnut St., is in the market for a large kw. 110 volt generator and a 110 volt d.c. motor.

Tenn., Nashville—R. Caldwell, c/o Caldwell & Co., 214 Union St., is having plans prepared for the construction of a 10 story office building on 4th and Union St. Estimated cost $500,000. Marr & Holman, Stahlman Bldg., Archt. Equipment detail not reported.

Tex., Dallas—A. T. Keith, Mayor, 715-16 Praetorian Bldg., is in the market for a crude oil engine, two centrifugal pumps, motor driven air compressor, steel tanks, hydrants and valves.

Tex., San Angelo—The San Angelo Water, Light & Power Co. is having plans prepared and will receive bids in 60 or 90 days for the construction of a filter plant, five units, 8 x 24, capacity 2,500,000 g.p.d. Estimated cost $50,000. Private plans.

W. Va., Charleston—The Charleston Market Co. is in the market for cold storage and refrigeration machinery for new $400,000 market here.

W. Va., Mason City—The Hutchison Coal Co. plans to rebuild its power plant, which was destroyed by fire. Estimated cost $70,000. The owner is in the market for power plant machinery boilers and equipment.

Wis., Fond du Lac—The Hotel Wisconsin Realty Co., 86 Michigan St., Milwaukee, is having plans prepared for the construction of an 8 story, 60 x 100 ft. hotel, (second unit). Estimated cost $300,000. M. Tullegren & Sons, 425 East Water St., Milwaukee, Archts. Equipment detail not reported.

Wis., Green Bay—The Green Bay Newspaper Co., 315 Cherry St., A. B. Trumbull, Secy., will receive bids until May 15, for the construction of a 2 story, 60 x 130 ft. newspaper and power plant. Estimated cost $100,000. Foeller & Schober, Archts., Green Bay.

Wis., Madison—The Hotel Wisconsin Realty Co., c/o W. Schroeder, 86 Michigan St., Milwaukee, is having plans prepared for the construction of an 8 story, 132 x 144 ft. hotel on West Washington and Fairfield Sts. Estimated cost $1,000,000. M. Tullegren & Sons, 425 East Water St., Milwaukee, Archts. Owner is in the market for refrigeration machinery, ice machine, laundry equipment, etc. Contract for excavation has been awarded. Noted April 17.

Wis., Madison—The Washington Bldg. Corp., 357 Washington Bldg., E. J. Ellis, Mgr., plans to build a 2 story heating plant. Estimated cost $80,000. Engineer or architect not selected.

Wis., Platteville—The Farmers Consumers Exch., A. Benedict, Mgr., is having plans prepared for the construction of a 2 story, 50 x 60 ft. cold storage and refrigeration plant. Estimated cost $75,000. Private plans. Owner is in the market for a 20 ton ice machine and refrigeration machinery. Noted Feb. 13.

Wis., Reeseville—The village, A. C. Buss, Clk., will soon receive bids for the construction of a waterworks system, including 1 deep well pump, motor driven, 50,000 gal. water tank, also 11,400 ft. sewers, septic tank.

Ont., Niagara Falls—The city is having plans prepared for the construction of an artificial ice skating and hockey arena, seating capacity about 3,500. Estimated cost $150,000. James, Proctor & Redfern, 36 Toronto St., Toronto, Engrs.

Ont., Norwood—The town, E. G. Laing, Purch. Agt., is in the market for centrifugal pumps, motors, valves, etc., for waterworks.

Ont., Toronto—The Manufacturers Life Inc. Co., Dominion Bank Bldg., plans the construction of a 8 story office building, including a steam heating system, elevators, etc. Estimated cost $1,500,000. Architect not selected.

CONTRACTS AWARDED

D. C., Wash.—The Cosmos Theater Co., 919 Pennsylvania Ave. N.W., awarded the contract for the construction of a theater and office building at 13th and E Sts., to the Longacre Engr. Co., 562 Fifth Ave., New York. Estimated cost $1,000,000. Equipment detail not reported.

Ill., Chicago—The Belmont Hotel Bldg., Corp., Room 1004, 400 North Michigan Ave., awarded the contract for the construction of a 12 story, 174 x 210 ft. hotel and apartment building at 3156-78 Sheridan Rd. to the H. McLennan Constr. Co., 400 North Michigan Ave. Estimated cost $1,250,000. Steam heating system will be installed.

Ia., Muscatine—The Bd. of Electric Light Trustees awarded the contract for the construction of a municipal electric power plant and distribution system to the Merkle Machinery and Contracting Co., 1733 Walnut St., Kansas City, Mo., $389,000. Noted April 24.

Kan., Wichita—O. S. Shirk, 805 Schweiter Bldg., awarded the contract for the construction of a 7 story, 100 x 140 ft. apartment hotel at 240 North Topeka St., to the G. H. Siedhoff Constr. Co., Broadview Hotel Bldg., Wichita. Estimated cost $400,000. Steam heating system, and 2 elevators, 1 freight and one passenger, will be installed. Noted Jan. 9.

Md., Baltimore—The Baltimore Ice Mfg. Co., c/o A. Beard, Pres., Hopkins Apartments, awarded the contract for the construction of a 2 story, 60 x 175 ft. ice manufacturing plant at 409-11 North Aisquith St. to J. J. Walsh & Son, Preston and Charles Sts., Baltimore. Estimated cost $150,000. Noted Mar. 20.

Minn., Minneapolis—The Northern States Power Co., 15 South 5th St., R. F. Peck, Vice-Pres., will build a 4 story, 60 x 77 x 100 ft. shop and office building at 2820 Marshall St., N. E., by day labor. Estimated cost $125,000. Equipment detail not reported.

Mo., St. Louis—The Fairmont Hotel Co., Chemical Bldg., awarded the contract for the construction of a 10 story, 147 x 216 ft. hotel on Euclid St. to N. Howard, Arcade Bldg., St. Louis. Estimated cost $500,000. Equipment detail not reported. Noted Mar. 13.

Mo., St. Louis—The St. Louis Dairy Co., 2030 Pine St., awarded the contract for the construction of a 3 story, 50 x 110 ft. ice cream and dairy plant to the Gamble Constr. Co., 620 Chestnut St., St. Louis. Estimated cost $200,000. The contract for the refrigeration machinery was awarded to the Vilter Ice Mchy. Co., 2735 Bennett Court, St. Louis.

N. J., Trenton—The Trenton Trust Co., Chancery and State Sts., awarded the general contract for the construction of a 14 story, 62 x 189 ft. building to the Karno-Smith Co., Broad St. Bank; structural iron to the Newton A. K. Bugbee Co., 206 East Hanover St., electrical work, J. A. Morrissey, 342 West 46th St., New York; heating contract not awarded. Estimated cost $1,000,000.

N. Y., Holley—The Holley Fruit & Cold Storage Co. awarded the contract for the construction of a 100 x 106 ft. cold storage plant to the Alexander, Shumway & Utz Co., Rochester, N. Y. Estimated cost $75,000. Owner is in the market for a 40 ft. compressor. Noted March 20.

N. Y., New York—The Bd. Educ., 500 Park Ave., awarded the contract for the construction of P. S. 66 on Jennings St. and P. S. 67 on East 179th St. to G. Dekempie, 30 Church St., New York; P. S. 70 on Weeks Ave. to T. Dwyer, Broadway and 216th St., New York. Estimated cost $1,019,950, $1,038,000 and $1,070,000 respectively. Equipment detail not reported. Noted Apr. 17.

N. Y., New York—H. B. Herts, Engr. and Archt., 331 Madison Ave., awarded the contract for the construction of an apartment building at 165 East 60th St. to the Fleischmann Constr. Co., 531 7th Ave., New York. Estimated cost $1,000,000. Owner's name withheld. Equipment detail not reported.

Ohio, Marysville—The Dept. Pub. Welware, 9th and Oak Sts., Columbus, awarded the contract for the construction of a 1 story, 48 x 60 ft. power plant at the Reformatory for Women, here, to E. Elford, 307 East Gay St., Columbus. Noted Apr. 17.

Tex., Texas City—The Texas Sugar Refining Co. awarded the contract for the construction of a sugar refinery to J. Monks & Son, 438 Bway., New York. Estimated cost $5,500,000. Equipment detail not reported. Noted Mar. 27.

Wis., Oconto Falls—The Falls Canning Co., awarded the contract for the construction of a 2 and 1 story 50 x 84 ft. and 30 x 32 ft. warehouse and boiler house to H. J. Selmer Co., McCartney Natl. Bank Bldg., Green Bay. Estimated cost $40,000. Private plans.

Wis., Sheboygan—The J. & W. Jung Co., 720 North 8th St., awarded the contract for the construction of a 1 story, 50 x 100 ft. refrigeration and central heating plant to the Verhulst Co., Inc., 2220 Calumet Dr. Estimated cost $50,000. W. C. Weeks, Engr., 720 Ontario Ave., will award contracts for steam heating plant, including boilers. Noted Jan. 16.

PRICES - MATERIALS - SUPPLIES

Electrical prices on following page are prices to the power plant by jobbers in the larger buying centers east of the Mississippi. Elsewhere the prices will be modified by increased freight charges and by local conditions.

SINCE LAST MONTH

Advances—Discounts reduced two points on both black and galvanized wrought-steel pipe on latest Pittsburgh basing card of April 19. Mill change reflected in higher warehouse prices. Structural steel shapes $3.64 at New York warehouses as against $3.44 per 100 lb. last month. Chrome brick $1 per net ton higher at Eastern shipping points. Steel and tinned rivets discounts down five and ten points, respectively, in New York warehouses. Following recent rapid advance in paint materials, white and red lead rose ½c. per lb., f.o.b. New York, during month. Armored cable, two and three conduit, advanced $5 per M. ft. Rise of 2 per cent in prices of conduit, elbows and couplings. Advances in cut-outs, flexible cord, brass shell sockets and rubber-covered copper wire, since last month.

Declines—Asbestos pipe and boiler covering discounts increased two to five points since April. Silica brick down 50c. per M. at Mt. Union, Pa. Babbitt metal slightly cheaper in Cleveland. Non-refillable N.E.C. fuses lower. Electric lamp prices reduced to point below pre-war level.

POWER-PLANT SUPPLIES

HOSE—

	Fire	50-Ft. Lengths
Underwriters' 2½-in., coupled		53c. per ft.
Common, 2½-in., 3-ply	$1.00 per ft. list less 40–5%	

	Air	
	First Grade	Second Grade
¾-in., 3-ply, per ft.	$0.33	$0.25

Steam—Discounts from List

First grade....40–5% Second grade....45% Third grade....50%

RUBBER BELTING—The following discounts from list apply to transmission rubber and duck belting:

Competition..... 60–10% Best grade..... 50–10–5%

LEATHER BELTING—List price per ply, 12-in. wide, per lin. ft. $2.88

Grade	Discount from list
Medium	30–10%
Heavy	20–5–2½%

RAWHIDE LACING—For cut, best grade, 40%, 2nd grade, 50%. For laces in sides, best, 48c. per sq.ft.; 2nd, 43c. Semi-tanned: cut, 40%; sides, 48c. per sq.ft.

PACKING—Prices per pound:

Rubber and duck for low-pressure steam, ⅛ in.	$0.90
Asbestos for high-pressure steam, ⅛ in.	1.70
Duck and rubber for piston packing	.90
Flax, regular	1.10
Flax, waterproofed	1.70
Compressed asbestos sheet	.80
Wire insertion asbestos sheet	1.30
Rubber sheet	.45
Rubber sheet, wire insertion	.70
Rubber sheet, duck insertion	.50
Rubber sheet, cloth insertion	.25
Asbestos packing, twisted or braided and graphited, for valve stems and stuffing boxes	1.30
Asbestos wick, ½- and 1-lb. balls	.50

PIPE AND BOILER COVERING—Discounts, New York warehouses, are as follows:

85% magnesia high pressure		45%
For low-pressure heating and return lines	4-ply	70%
	3-ply	72½%
	2-ply	75%

PORTLAND CEMENT—New York, $2.70@ $2.80 per bbl. without bags, in cargo lots delivered on job. Bag charge of 40c. per bbl.

STRUCTURAL STEEL—New York delivered price, 3 to 15-in. beams and channels and 3 to 6-in. angles, tees, and plates, all $3.64 per 100 lb.

COTTON WASTE—The following prices are in cents per pound:

	New York Current	Cleveland	Chicago
White	10.00@13.00	15.00	11.25
Colored	7.00@12.00	12.00	8.00

WIPING CLOTHS—Jobbers' prices, in cents per lb., as follows:

	13¼ x 13¼	13¼ x 20½
Chicago	10.00	13.00
New York	11.00@11.25	
Cleveland	$36.00 per M.	$52.00 per M.

LINSEED OIL—These prices are per gallon:

	New York	Cleveland	Chicago
Raw in barrels (5 bbl. lots)	$1.16	$1.10	$1.25

WHITE AND RED LEAD—Base price in cents per pound:

	Red				White			
	Current		1 Year Ago		Current		1 Yr. Ago	
	Dry	In Oil	Dry	In Oil	Dry or In Oil	In Oil	Dry or In Oil	In Oil
100-lb. keg	14.50	16.00	12.25	13.75	14.50		12.25	
25- and 50-lb. keg	14.75	16.25	12.50	14.00	14.75		12.50	
12½-lb. keg	15.00	16.50	12.75	14.25	15.00		12.75	
5-lb. cans	17.50	19.00	15.25	16.75	17.50		15.25	
1-lb. cans	19.50	21.00	17.25	18.75	19.50		17.25	

RIVETS—The following quotations are allowed for fair-sized orders from warehouse:

	New York	Cleveland	Chicago
Steel ⅛ and smaller	40%	50–10%	60%
Tinned	40%	50–10%	4½c. per lb. net

Structural rivets, ½, ⅝, 1 in. diameter by 2 in. to 5 in. sell as follows per 100 lb.

New York.....$4.00 Chicago.....$3.75 Pittsburgh..$3.25@3.50

Boiler rivets, same sizes:

New York.....$4.10 Chicago..... $3.85 Pittsburgh..$3.35@3.60

REFRACTORIES—Prices in carlots f.o.b. plant:

Chrome brick, eastern shipping points	net ton	$52@55
Chrome cement, 40@ 50% Cr_2O_3, in bulk	net ton	27@32
Chrome cement, 40@50% Cr_2O_3, in sacks	net ton	31@36
Magnesite brick: 9-in. shapes	net ton	65@68
Magnesite brick: 9-in. arches, wedges and keys	net ton	71.50@74 80
Magnesite brick: Soaps and spits	per ton	92.00@96.20
Silica brick: Chicago district	per M	52.00
Silica brick: Birmingham, Ala.	per M	51.00
Silica brick: Mt. Union, Pa.	per M	46.50
Clay brick, 1st quality, 9 in. shapes, Pennsylvania	per M	48@55
Clay brick, 1st quality, 9 in. shapes, Ohio	per M	45@47
Clay brick, 1st quality, 9 in. shapes, Kentucky	per M	45@47
Clay brick, 2nd quality, 9 in. shapes, Pennsylvania	per M	43@50
Clay brick, 2nd quality, 9 in. shapes, Ohio	per M	40@43
Clay brick, 2nd quality, 9 in. shapes, Kentucky	per M	40@45
Chrome ore crude, 40@50%	net ton	25.50
Magnesite dead burn	net ton	39.50

BABBITT METAL—Warehouse prices in cents per pound:

	New York	Cleveland	Chicago
(83% tin)	65.00	56.00	36.00
(35% tin)	40.00	18.50	.....

COLD FINISHED STEEL—Warehouse prices are as follows:

	New York	Cleveland	Chicago
Round shafting and screw stock, per 100 lb. base	$4.20	$3.90	$4.30
Flats, square and hexagons, per 100 lb. base	4.70	4.40	4.80

BOILER SPECIALTIES—F. o. b. New York or Jersey City, discounts on list:

	Current
Copper ferrules	70%
Boiler flanges	60-10%
Boiler stay bolts	60%
Boiler patch bolts	10%
Boiler fitting-up bolts	45%
Pressed steel boiler lugs	10%
Pressed steel boiler hangers	10%

WROUGHT PIPE—The following discounts are to jobbers for carload lots on the latest Pittsburgh basing card:

BUTT WELD

Inches	Steel Black	Steel Galv.	Inches	Iron Black	Iron Galv.
1 to 3	62	50½	¼ to 1¼	30	13

LAP WELD

2	55	43½	2	23	7
2½ to 6	59	47½	2½ to 4	26	11
7 and 8	56	43½	4½ to 6	28	13
9 and 10	54	41½	7 to 12	26	11
11 and 12	53	40½			

BUTT WELD, EXTRA STRONG, PLAIN ENDS

1 to 1¼	60	49½	¼ to 1¼	30	14
2 to 3	61	50½			

LAP WELD, EXTRA STRONG, PLAIN ENDS

2	53	42½	2	23	9
2½ to 4	57	46½	2½ to 4	29	15
4½ to 6	56	45½	4½ to 6	28	14
7 and 8	52	39½	7 and 8	21	7
9 and 10	45	32½	9 to 12	16	2
11 and 12	44	31½			

BOILER TUBES—Following are prices in New York warehouse of tubes manufactured according to specifications of the American Society of Mechanical Engineers:

Size	Lapweld Steel	C. C. Iron	Seamless Steel
1	.....	.....	$0.23
1¼	.....	.....	.25
1½	.....	.....	.22
1¾	$0.26	$0.29	.23
2	.20	.25	.21
2¼	.23	.29	.23
2½	.24	.33	.28
2¾	.....	.....	.....
3	.30	.41	.32
3¼	.32	.47	.35
3½	.34	.51	.37
4	.43	.64	.47

Tubes 2½ in. diameter, or smaller, over 18 ft. long, 10 per cent extra.

These prices are net per lineal foot based on stock lengths. If cut to special engths, billing will be based on the entire stock lengths.

In addition to the above, standard cutting charges are as follows:

1¼ in. to 2 in. diameter, 5c. per cut. 2¾ in. diameter, 7c. per cut
2½ in. diameter, 6c. per cut. 3 in. diameter, 9c. per cut
3½ in. to 4 in. diameter, 10c.

ELECTRICAL SUPPLIES

ARMORED CABLE—Price per 1,000 ft.—5 per cent 10 days.

B. & S. Size	Two Cond. M Ft.	Three Cond. M Ft.	Two Cond. Lead M Ft.	Three Cond. Lead M Ft.
No. 14 solid	$ 52.00 (net)	$84.00 (net)	$164.00	$210.00
No. 12 solid	135.00	170.00	225.00	265.00
No. 10 solid	185.00	235.00	275.00	325.00
No. 8 stranded	285.00	375.00	520.00	500.00
No. 6 stranded	400.00	500.00	560.00	.....

From the above lists discounts are:		Lead Covered
Less than coil lots	20%	10%
Coils to 1,000 ft.	30%	20%
1,000 to 5,000 ft.	35%	20-5%
5,000 ft. and over	40%	25%

CONDUIT, Price per 1,000 ft.; **ELBOWS AND COUPLINGS**, Per 100 pieces, f. o. b. New York, with 10-day discount of 5 per cent.

Size In.	Conduit Black Per M	Conduit Galvanized Per M	Elbows Black Per C	Elbows Galvanized Per C	Couplings Black Per C	Couplings Galvanized Per C
¼	$61.18	$65.37	$10.15	$11.32	$5.67	$6.09
⅜	76.71	83.75	13.36	14.89	8.09	8.71
½	109.93	120.33	19.77	22.04	10.52	11.32
¾	148.73	162.81	26.34	29.10	14.63	15.67
1	177.89	194.15	35.12	38.80	18.07	19.35
1¼	239.27	261.91	64.40	71.13	24.10	25.81
1½	378.30	414.10	105.38	116.40	34.43	36.88
2	494.71	541.52	281.03	310.40	51.75	55.32
2½	623.09	679.40	700.60	685.48	68.87	73.76
3	750.47	827.17	717.21	792.18	86.08	92.20

CONDUIT BODIES AND FITTINGS—Black or galvanized.

	Less than $10 list	$10 list to $100	$100 list and over
Standard package	10%	20%	28%
Less than standard package	5%	10%	20%

CUT-OUTS—Following are net prices each in standard-package quantities:

CUT-OUTS, PLUG

S. P. M. L.	$0.10	D. P. D. B.	$0.32
D. P. M. L.	.16	T. P. S. B.	.30
T. P. M. L.	.24	T. P. D. B.	.42
D. P. S. B.	.18		

CUT-OUTS, N. E. C. FUSE

	0-30 Amp.	31-60 Amp.	60-100 Amp.
D. P. M. L.	$0.39	$1.00	$2.10
T. P. M. L.	.60	1.47	2.80
D. P. S. B.	.49	1.26	.....
T. P. S. B.	.92	2.10	.....
D. P. D. B.	.89	2.46	.....
T. P. D. B.	1.57	4.20	.....
T. P. to D. P. D. B.	1.05	2.88	.....

FLEXIBLE CORD—Price per 1,000 ft. in coils of 250 ft.:

No. 18 cotton reinforced heavy	$23.00
No. 16 cotton reinforced heavy	29.00
No. 18 cotton reinforced light	19.50
No. 16 cotton reinforced light	22.50
No. 18 cotton Canvasite cord	17.00
No. 16 cotton Canvasite cord	19.25

NATIONAL ELECTRIC CODE FUSES, NON-REFILLABLE—

250-Volt	Std. Pkg.	List	600-Volt	Std. Pkg.	List
3-amp. to 30-amp.	100	$.20	3-amp. to 30-amp.	100	$0.40
35-amp. to 60-amp.	100	0.30	35-amp. to 60-amp.	100	.60
61-amp. to 100-amp.	50	.90	65-amp. to 100-amp.	50	1.50
101-amp. to 200-amp.	25	2.00	110-amp. to 200-amp.	25	2.50
201-amp. to 400-amp.	25	3.60	225-amp. to 400-amp.	25	5.50
401-amp. to 600-amp.	10	5.50	450-amp. to 600-amp.	10	8.00

Discount: Less 1-5th standard package, 55%; 1-5th to standard package, 60%; standard package, 65%.

RENEWABLE FUSES, ENCLOSED—List price each:

Sizes	250-Volt List-Price	600-Volt List-Price	Std. Pkg. Quantity	Carton Quantity
1 to 30-amp.	$0.50	$1.10	100	10
35 to 60-amp.	1.00	1.25	100	10
65 to 100-amp.	2.00	3.00	50	5
110 to 200-amp.	4.00	5.00	25	5
225 to 400-amp.	7.50	11.00	25	1
450 to 600-amp.	11.00	16.00	10	1
450 to 600-amp.	11.00	16.00	10	1

REFILLS—

1 to 30	$0.30 ea.	$0.05	100	100
35 to 60	.05 ea.	.06	100	100
65 to 100	.10 ea.	.10	50	50
110 to 200	.15 ea.	.15	25	50
225 to 400	.30 ea.	.30	25	25
450 to 600	.60 ea.	.60	10	10

Discount Without Contract—Fuses:	
Broken carton	5%
Unbroken carton but less than std. pkg.	22%
Std. pkg.	40%
Discount Without Contract—Renewals:	
Less than std. pkg.	Net list
Standard package	40%
Discount With Contract—Fuses:	
Broken cartons	10%
Unbroken cartons but less than standard package	26%
Standard package	42%
Discount With Contract—Renewals:	
Less standard package	Net list
Standard package	42%

FUSE PLUGS, MICA CAP—

0-30 ampere, standard package (500)	$2.70
0-30 ampere, less than standard package	3.00

LAMPS—Below are present quotations in less than standard package quantities:

Straight-Side Bulbs Mazda B— Watts	Plain	Frosted	No. in Package	Pear-Shaped Bulbs or Bowl Enameled Mazda C— Watts	Clear	Frosted	No. in Package
10	$0.32	$0.37	120	75	$0.55	$0.60	60
15	.32	.37	120	100	.70	.75	24
25	.32	.37	120	150	.90	.95	24
40	.32	.37	120	200	1.15	1.25	24
50	.32	.37	120	300	1.75	1.85	24
60	.37	.42	120				
				500	2.50	2.65	12
				750	3.75	4.00	8
				1,000	4.25	4.50	8

Standard quantities are subject to discount of 10% from list. Annual contracts ranging from $150 to $300,000 net allow a discount of 17 to 40% from list.

PLUGS, ATTACHMENT—

	Each
Porcelain separable attachment plug	$0.19
Composition 2-piece attachment plug	.27
Swivel attachment plug	.12

RUBBER-COVERED COPPER WIRE—Per 1000 ft. f. o. b. New York.

No.	Solid Single Braid	Solid Double Braid	Stranded, Double Braid	Duplex
14	$ 7.95	$10.30	$12.30	$19.80
12	10.50	13.40	14.80	25.50
10	14.40	16.70	19.00	33.00
8	20.10	22.58	24.90	42.10
6	.....	.....	37.60	.....
4	.....	.....	50.78	.....
2	.....	.....	14.80	.....
1	.....	.....	101.30	.....
0	.....	.....	122.00	.....
00	.....	.....	146.10	.....
000	.....	.....	176.00	.....
0000	.....	.....	212.00	.....

SOCKETS, BRASS SHELL—

½ In. or Pendant Cap Key Each	Keyless Each	Pull Each	¾ In. Cap Key Each	Keyless Each	Pull Each
$0.33	$0.30	$0.60	$0.39	$0.36	$0.66

Less 1-5th standard package	20%
1-5th to standard package	25%
Standard package	35%

WIRING SUPPLIES—

Friction tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c. lb.
Rubber tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c. lb.
Wire solder, less 100 lb. 27c. lb., 100 lb. lots	29c. lb.
Soldering paste, 2 oz. cans	$1.00 doz.

ENCLOSED SWITCHES, KNIFE—Externally operated, 250 d.c. or a.c., N.E.C.

TYPE "C" FUSED BOTTOM

Size, Amp.	Double Pole, Each	Three Pole, Each	Four Pole, Each
30	$4.50	$6.00	$7.25
60	7.50	8.25	10.50
100	10.50	13.00	22.50
200	16.00	20.00	36.00

Discounts:	
Less than $25.00 list value	25%
$25 to $50 list value	30%
$50 list value or over	35%